TRANSACTIONS

American Society of Heating and Air-Conditioning Engineers

VOLUME 64

Sixty-fourth Annual Meeting
Pittsburgh, Pa.
January 27–29, 1958

Semi-Annual Meeting
Minneapolis, Minn.
June 23–25, 1958

Published by the Society at the Office of the Executive Secretary
62 Worth St., New York 13, N. Y.
Printed in U.S.A.

Printed and bound by

The Horn-Shafer Co.
Baltimore, Md.
U.S.A.

CONTENTS

Officers and Council—1958

American Society of Heating and Air-Conditioning Engineers

Headquarters: 62 Worth St., New York 13, N. Y.
(Tel.: BArclay 7-6262)

OFFICERS

President	E. R. Queer
First Vice President	A. J. Hess
Second Vice President	Walter A. Grant
Treasurer	J. H. Fox
Executive Secretary	A. V. Hutchinson

COUNCIL

E. R. Queer, *Chairman* A. J. Hess, *Vice Chairman*

Three Years	*Two Years*	*One Year*
F. H. Faust	W. G. Hole	A. B. Algren
Fred Janssen	P. J. Marschall	John Everetts, Jr.
J. W. May	W. O. Stewart	P. B. Gordon
G. B. Priester	P. N. Vinther	J. N. Livermore
		J. H. Ross

REGIONAL DIRECTORS

Chairman: Walter A. Grant

Region 1	F. H. Faust
Region 2	P. J. Marschall
Region 3	Fred Janssen
Region 4	W. O. Stewart
Region 5	J. W. May
Region 6	P. N. Vinther
Region 7	W. G. Hole

STAFF

Technical Secretary	Carl H. Flink
Editor	C. H. B. Hotchkiss
Public Relations	W. M. Vidulich
Assistant Secretary	Fred W. Hofmann
Technical Assistant	Carl W. MacPhee

ADVISORY BOARD

P. B. Gordon, *Chairman*; Lester T. Avery, M. F. Blankin, S. E. Dibble, S. H. Downs, E. O. Eastwood, W. L. Fleisher, H. P. Gant, H. M. Hart, John E. Haines, C. V. Haynes, L. N. Hunter, John W. James, Samuel R. Lewis, Alfred J. Offner, F. B. Rowley, L. E. Seeley, A. E. Stacey, Jr., Reg. F. Taylor, G. L. Tuve, and A. C. Willard.

Council Committees

Executive: A. J. Hess, *Chairman*; J. H. Fox, P. B. Gordon, Walter A. Grant, E. R. Queer.

Finance: J. N. Livermore, *Chairman*; John Everetts, Jr., F. H. Faust, P. J. Marschall, J. H. Fox (*Ex Officio*).

Membership: J. H. Ross, *Chairman*; A. B. Algren, F. H. Bridgers, Fred Janssen, W. O. Stewart.

Program and Papers: John Everetts, Jr., *Chairman*; W. G. Hole, J. W. May, G. B. Priester, P. N. Vinther.

Regions Central: Walter A. Grant, *Chairman*; F. H. Faust (1); P. J. Marschall (2); Fred Janssen (3); W. O. Stewart (4); J. W. May (5); P. N. Vinther (6); W. G. Hole (7).

General Committees

Admission and Advancement: Roswell Farnham, *Chairman* (*one year*); C. S. Koehler (*two years*); P. R. Achenbach (*three years*).

Publication: J. W. May, *Chairman* (*one year*); H. M. Hendrickson (*two years*); W. L. Ross (*three years*).

Charter and By-Laws: J. C. Fitts, *Chairman* (*one year*); J. S. Locke (*two years*); G. B. Supple (*three years*).

F. Paul Anderson (Honors and Awards): A. J. Hess, *Chairman*; John W. James, C. H. Pesterfield, N. H. Peterson, R. A. Sherman, E. F. Snyder, Jr. (*Ex Officio*).

Guide: H. W. Alyea, *Chairman*; W. P. Chapman, R. C. Chewning, J. R. Duncan, R. A. Gonzalez, N. B. Hutcheon, M. W. Keyes, R. L. Maher, D. W. Nelson, E. F. Snyder, Jr. (*Ex Officio*).

Chapters Regional Committees

Regional Areas	*Chapter*	*Member*	*Alternate*
Region 1—Regional Director: F. H. Faust			
Conn.	Connecticut (New Haven)	Walter Heywood, Jr.	J. D. Pierce
Del., Maine			
Mass.	Massachusetts (Boston)	E. L. Blair	R. F. Curry
Mass.	Western Massachusetts (Springfield)	F. A. Ferraro	C. R. Munn
N. H.			
N. J.	North Jersey (Newark)	S. H. Nitzberg	L. G. Huggins
N. Y.	Central New York (Syracuse)	L. A. Childs	S. F. Gilman
N. Y.	Empire State Capital (Albany)	H. F. Kruger	R. B. Taylor
N. Y.	Long Island	H. J. Campbell, Jr.	C. M. Alston
N. Y.	New York	J. B. Hewett	C. R. Hiers
N. Y.	Western New York (Buffalo)	C. W. Stone	J. P. Guerra
Pa.	Philadelphia	E. K. Wagner	C. J. Forve
Pa.	Pittsburgh	E. H. Riesmeyer, Jr.	E. J. Busch
Pa.	Johnstown	K. O. Schlentner	J. K. Thornton
R. I., Vt.			
Region 2—Regional Director: P. J. Marschall			
Ill.	Illinois (Chicago)	H. E. Anderson	Herbert Kreisman
Ind.	Indiana (Indianapolis)	A. O. Roche, Jr.	G. W. Vogel
Mich.	Michigan (Detroit)	D. S. Falk	K. J. Wagoner
Mich.	Western Michigan (Grand Rapids)	B. J. Walter	R. J. Waalkes
Ohio	Central Ohio (Columbus)	W. A. Schoonover	T. R. Walker
Ohio	Cincinnati	T. D. Reiley	C. P. Krantz
Ohio	Miami Valley (Dayton)	L. P. Brehm	R. B. Walcott
Ohio	Northern Ohio (Cleveland)	J. L. Frissee	R. A. Urban
Ohio	Toledo	J. F. Guest	Robert Greenwald
Wis.	Wisconsin (Milwaukee)	R. D. Rodwell	A. A. Stutheit
Region 3—Regional Director: Fred Janssen			
Colo.	Rocky Mountain (Denver)	A. S. Widdowfield	D. D. Pearsall
Iowa	Iowa (Des Moines)	R. S. Stover	R. T. Howard, Jr.
Iowa	Illinois-Iowa (Moline, Ill.)	W. W. Dodge	D. H. Ninow
Kan.	Kansas (Wichita)	T. L. Roberts	R. F. Jamison
Minn.	Minnesota (Minneapolis)	F. G. Vogt	J. H. Jester
Mo	Kansas City	J. C. Fasnacht	L. A. Heaven
Mo.	St. Louis	J. J. Blackmore	N. J. Hubbuch
Mont.			
Neb.	Nebraska (Omaha)	E. N. Seiler	F. P. Manchester, Jr.
N. D., S. D., Wyo.			
Region 4—Regional Director: W. O. Stewart			
Ariz.	Arizona (Phoenix)	E. P. Boothroyd	F. W. Gabbard
Calif.	Golden Gate (San Francisco)	Herb Duncan	J. I. Sprott
Calif.	Sacramento Valley (Sacramento)	M. J. Delavan	J. A. White
Calif.	Southern California (Los Angeles)	W. J. Biggar	A. Z. Levine
Idaho, Nev.			
Ore.	Oregon (Portland)	Keith Kruchbeck	W. B. Hayes
Utah	Utah (Salt Lake City)	L. K. Irvine	C. E. Ferguson
Wash.	Puget Sound (Seattle)	M. W. McKinstry	H. M. Hendrickson
Wash.	Inland Empire (Spokane)	R. D. Nevers	J. A. Doyle

General Committees—1958 (*Continued*)

Region 5—Regional Director: J. W. May

Ala.			
Fla.	Jacksonville	W. H. Dowling, Jr.	H. A. Holborn
Ga.	Atlanta	J. S. Edgar	W. P. West
Ga.	Savannah	R. L. Reilley	S. H. Ball
Ky.	Bluegrass (Louisville)	O. J. Hill	H. L. Carr
La.	Delta (New Orleans)	J. H. Maloney	W. B. Moses
La.	Baton Rouge	A. W. Holland	W. J. Le Blanc
Md.	Baltimore	R. E. Dressell	F. M. Hewitt
Miss.	Mississippi (Jackson)	L. H. Smith	R. E. Vernon
N. C.	Northern Piedmont (Greensboro)	J. E. Hart	G. B. Rottman
N. C.	Southern Piedmont (Charlotte)	W. P. Wells	J. R. Clark
S. C.	South Carolina (Columbia)	B.A. Leppard	H. J. Haar, Jr.
Tenn.	Memphis	G. B. Ramsey, Jr.	J. G. Coleman
Va.	Virginia (Norfolk)	D. L. Gusler	J. A. Hoffman
D. C.	Washington, D. C.	H. W. Rush	J. W. Morgan
W. Va.			

Region 6—Regional Director: P. N. Vinther

Ark.	Arkansas (Little Rock)	J. W. Thompson	W. A. Bullard
La.	Shreveport	J. S. Malahy, Jr.	F. H. Spaulding, Jr.
N. Mex.	New Mexico (Albuquerque)	F. H. Bridgers	E. T. Krytko
Okla.	Northeastern Oklahoma (Tulsa)	J. E. Tumilty	G. A. McCune
Okla.	Oklahoma (Oklahoma City)	F. R. Denham	W. R. Johnson
Texas	Austin	J. L. Rea, Jr.	W. H. Luedecke
Texas	El Paso	G. H. Jackson	F. A. Huchet, Jr.
Texas	North Texas (Dallas)	G. H. Meffert	O. H. Mehl
Texas	South Texas (Houston)	H. D. McMillan	E. B. Appling
Texas	Southwest Texas (San Antonio)	F. S. Dalton	D. T. Kern
Texas	West Texas (Lubbock)	L. F. Davidson	R. A. Mixon
Texas	Fort Worth	T. B. Romine, Jr.	J. K. Mattox, Jr.

Region 7—Regional Director: W. G. Hole

B. C.	British Columbia (Vancouver)	S. W. Welsh	A. C. Martin
Man.	Manitoba (Winnipeg)	H. F. Randall	A. K. Piercy
Que.	Montreal	D. J. McIntyre	Arthur de Breyne
Que.	Châpitre de la Ville de Québec	Jean Veilleux	P. J. Lamarche
Que.	Ontario (Toronto)	John Thompson	R. A. Ritchie
Ont.	Ottawa Valley (Ottawa)	Jacob Klassen	A. H. Hargreaves
Ont.	Northern Alberta (Edmonton)	R. G. Proudfoot	G. N. Campbell
Alta.	Southern Alberta (Calgary)	H. W. Klassen	L. D. Ontonio

Committee on Research—1958

ASHAE Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio
(Tel.: UTah 1-6161)

E. F. Snyder, Jr., *Chairman* — R. C. Jordan, *Vice Chairman*
B. H. Jennings, *Director of Research* — C. M. Humphreys, *Assistant Director*

Three Years	*Two Years*	*One Year*
P. R. Achenbach	D. D'Eustachio	F. K. Hick, M.D.
S. F. Gilman	E. S. Howarth	R. C. Jordan
N. B. Hutcheon	H. R. Limbacher	H. A. Lockhart
R. M. Stern	E. F. Snyder, Jr.	Arthur Nutting
H. E. Ziel	J. I. Trimble	

Research Executive: E. F. Snyder, Jr., *Chairman*; R. C. Jordan, *Vice Chairman*; D. D'Eustachio, N. B. Hutcheon, H. L. Lockhart.

Long-Range Research Program: R. C. Jordan, *Chairman*; H. A. Lockhart, *Vice Chairman*; N. B. Hutcheon, Arthur Nutting, J. I. Trimble.

Technical Advisory Committees—1958

Air Cleaning: W. C. L. Hemeon, *Chairman*; Arthur Nutting*, *Vice Chairman*; P. R. Achenbach, C. W. Coblentz, S. F. Duncan, O. C. Eliason, P. E. Engle, B. L. Evans, M. W. Keyes, R. C. Lower, C. H. Meyer, C. S. Stephens, J. R. Swanton, Jr.

* Member of Committee on Research.

Air Distribution: P. A. Argentieri, *Chairman*; H. R. Limbacher*, *Vice Chairman*; J. V. Borry, J. C. Davidson, F. B. Holgate, W. O. Huebner, W. W. Kennedy, W. B. Kirk, Alfred Koestel, E. J. Kurek, R. D. Madison, R. G. Nevins, H. B. Nottage, H. M. Patrick, L. R. Phillips, N. S. Shataloff, H. E. Straub, R. D. Tutt, Fred Vogt, Hrant Yousoufian.

Combustion: M. W. McRae, *Chairman*; H. R. Limbacher*, *Vice Chairman*; J. J. Boland, K. T. Davis, W. B. Kirk, D. W. Locklin, A. A. Marks, E. M. Mittendorff, W. M. Myler, Jr., C. H. Neiman, Jr., E. O. Olson, W. W. Popyk, R. W. Sage, D. L. Wooden.

Evaporative Cooling: S. F. Duncan, *Chairman*; R. M. Stern*, *Vice Chairman*; R. S. Ash, D. R. Baker, H. S. Dutcher, Leo Hungerford, L. C. Pellegrini, C. W. Pollock, C. B. Rowe, W. T. Smith, B. H. Spurlock, Jr., J. R. Watt.

Heat Pump: P. R. Achenbach*, *Chairman*; R. C. Jordan*, *Vice Chairman*; E. R. Ambrose, D. N. Berks, G. L. Biehn, Warren Cheatham, R. C. Chewning, S. F. Graziano, W. L. McGrath, W. E. Nanes.

Heat Transfer Through Fenestration: D. D'Eustachio*, *Chairman*; R. C. Jordan*, *Vice Chairman*; T. C. Carson, O. D. Englehart, W. B. Ewing, C. W. Kelly, J. B. Leavy, E. C. Miles, Bruno Morabito, F. W. Mowrey, G. R. Munger, O. L. Pierson, D. J. Vild, O. F. Wenzler, C. J. Youngblood.

Heating and Air Conditioning Loads: H. T. Gilkey, *Chairman*; P. R. Achenbach*, *Vice Chairman*; R. T. Baum, R. G. Chapman, C. W. Coblentz, R. A. Gonzalez, W. G. Kane, T. C. Min, G. R. Munger, I. A. Naman, T. F. Rockwell, A. M. Simpson, D. B. Turkington, A. Grant Wilson, W. R. Yeary.

Hot Water and Steam Heating: A. O. Roche, Jr., *Chairman*; H. E. Ziel*, *Vice Chairman*; G. F. Carlson, J. R. Carroll, J. W. James, D. L. Lindsay, R. B. Luhnow, Jr., E. M. Mittendorff, J. D. Pierce, K. O. Schlentner, L. E. Seeley, R. M. Stern*, E. R. Teske, N. E. Westphal.

Industrial Environment: Arthur Nutting*, *Chairman*; J. H. Clarke, *Vice Chairman*; Lucien Brouha, F. N. Calhoon, K. J. Caplan, C. S. Cave, A. S. Gates, Jr., Robert Greenwald, T. F. Hatch, E. C. Hungate, J. M. Kane, H. P. Kaulfuss, R. L. Kuehner, B. R. Small, R. P. Warren.

Insulation: G. R. Munger, *Chairman*; H. A. Lockhart*, *Vice Chairman*; J. M. Barnhart, W. J. Binder, H. C. Brown, Jr., C. W. Foster, D. A. Huebner, W. T. Irwin, F. A. Joy, R. A. LaCrosse, R. J. O'Heir, F. E. Parsons, E. L. Perrine, C. F. Pridmore, K. M. Ritchie, Laurence Shuman, W. C. Turner, J. D. Ueber.

Odors: H. L. Barnebey, *Chairman*; J. I. Trimble*, *Vice Chairman*; C. J. D'Angio, C. F. Kayan, R. L. Kuehner, E. H. Lindemann, Lawrence Macrow, Douglas E. Melzard, P. J. Messer, Sherman Ross, S. B. Smith, L. A. Staebler, R. D. Touton, Warren Viessman.

Physiological Research and Human Comfort: John Everetts, Jr., *Chairman*; F. K. Hick*, M.D., *Vice Chairman*; Thomas Bedford, H. S. Belding, Lucien Brouha, M. K. Fahnestock, T. F. Hatch, L. N. Hunter, N. B. Hutcheon*, John W. James, P. J. Marschall, R. G. Nevins, P. W. Wyckoff, C. P. Yaglou.

Plant and Animal Husbandry: H. A. Lockhart*, *Chairman*; T. E. Bond, M. K. Fahnestock, A. J. Hess, C. F. Kelly, G. L. Nelson, R. E. Stewart, H. J. Thompson, H. E. Ziel*.

Solar Energy Utilization: R. C. Jordan*, *Chairman*; M. L. Ghai, *Vice Chairman*; E. R. Ambrose, F. R. Ellenberger, N. B. Hutcheon*, M. L. Khanna, G. O. G. Löf, Gunnar Pleijel, Maria Telkes, J. L. Threlkeld, G. T. Ward, L. F. Yissar.

---

* Member of Committee on Research.

Sorption: G. C. F. Asker, *Chairman*; E. S. Howarth*, *Vice Chairman*; H. L. Barnebey, W. P. Cummings, W. E. Emley, Jr., R. J. Getty, R. R. Goll, T. E. Gravenstreter, Anthony Hass, G. A. Kelley, E. R. McLaughlin, R. M. Milton, H. E. Parker, W. L. Ross, G. L. Simpson.

Sound and Vibration Control: A. F. Hubbard, *Chairman*; D. D'Eustachio*, *Vice Chairman*; C. M. Ashley, Sydney Baken, J. B. Chaddock, R. A. Gerlitz, J. B. Graham, R. N. Hamme, John Harker, C. R. Hiers, F. B. Holgate, F. D. Joesting, D. H. Krans, R. D. Lemmerman, R. E. Parker, G. J. Sanders, R. M. Towne, R. J. Wells.

Thermal Circuits: S. F. Gilman*, *Chairman*; Harry Buchberg, *Vice Chairman*; N. R. Gay, C. F. Kayan, C. B. Monk, Jr., L. W. Nelson, H. B. Nottage, K. R. Solvason.

Weather Data: W. L. Holladay, *Chairman*; P. R. Achenbach*, James Bricker, D. T. Donovan, John Everetts, Jr., D. L. Fiske, J. B. Graham, W. S. Harris, W. J. Radle, W. T. Smith, H. C. S. Thom.

•

Public Relations: E. K. Wagner, *Chairman*; T. V. Johnson and A. C. Lagow, Advisory Members; E. J. Morris, R. S. Stover, J. H. Fox.

Standards: P. N. Vinther, *Chairman*; A. B. Algren; K. T. Davis; Albert Giannini; Fred Janssen; G. B. Priester.

ASHAE Committee on Minimum Outdoor Air Requirements for Air Conditioned Spaces: Albert Giannini, *Chairman*; C. B. Gamble, R. A. Gonzalez, W. J. McConnell, R. J. Salinger, H. G. Strong.

ASHAE Committee on Revision of Standard for Testing and Rating Return Line Low Vacuum Heating Pumps: W. M. Wallace II, *Chairman*; H. E. Adams, H. S. Bean, A. H. Church, L. M. Nechine, E. G. Keller, A. O. Roche, Jr., D. H. Whittington.

ASHAE Committee for Review of Standard Test Code for Air Moving Devices: S. Konzo, *Chairman*; L. G. Seigel, G. L. Tuve, D. D. Wile.

ASHAE Committee on Revision of Code for Testing and Rating Unit Ventilators: J. W. McElgin, *Chairman*; L. A. Cherry, A. F. Hubbard, D. H. Kranz, L. G. Miller.

ASHAE Committee on Guidance for Members in Preparation of Local Codes: H. G. Strong, *Chairman*; J. R. Chamberlain, K. T. Davis, C. L. Grandstaff, W. G. Martin, Jr., H. B. Nottage, C. H. Randolph, A. L. Wales.

ASHAE Committee on Test Standard for Testing and Rating Gravity Type Roof Ventilators: Linn Helander, *Chairman*; G. C. Breidert, F. N. Calhoon, R. B. Engdahl, W. V. Hukill, J. P. Johnson, K. E. Robinson, F. B. Rowley, L. J. Shumaker.

ASHAE Committee on Test Standard for Evaluating Air Cleaning Devices for General Ventilation: P. R. Achenbach, *Chairman*; W. C. L. Hemeon, R. F. Logsdon, P. E. McNall, R. J. Salinger.

ASHAE Committee on Test Standard for Sorbent Type Dehumidifiers: G. C. F. Asker, *Chairman*; E. S. Howarth, H. L. Barnebey, W. P. Cummings, W. E. Emley, Jr., R. J. Getty, R. R. Goll, T. E. Gravenstreter, Anthony Hass, G. A. Kelley, E. R. McLaughlin, R. M. Milton, H. E. Parker, W. L. Ross, G. L. Simpson.

ASHAE Committee on Code for Testing Heavy Duty Furnaces and Direct-Fired Unit Heaters: E. K. Campbell, *Chairman*; Bowen Campbell, A. P. Loeb, C. J. Low, W. J. MaGirl, B. F. McLouth, F. L. Meyer, L. G. Miller, A. A. Olson, H. A. Pietsch, O. J. Ress, H. A. Soper.

* Member of Committee on Research.

General Committees—1958 (*Continued*)

ASHAE Committee on Code for Testing and Rating Evaporative Coolers: R. S. Ash, *Chairman*; S. F. Duncan, H. S. Dutcher, E. F. Gritton, Leo Hungerford, C. B. Rowe, W. T. Smith, J. R. Watt.

Special Committees—1958

ASHAE-AIA Committee on Cooperation: John E. Haines, *Co-Chairman*; **ASHAE** Representatives—John Everetts, Jr., H. C. Hoffman, G. A. Linskie, R. J. Salinger; **AIA** Representatives—Angelo R. Clas, *Co-Chairman*; S. T. Franklin, Sr., George S. Idell, R. E. Langdon, Jr., James B. Newman.

ASHAE-ASRE Committee on Cooperation: Representatives—**ASHAE** C. B. Gamble, W. G. Hole, L. N. Hunter, P. J. Marschall, E. R. Queer. **ASRE** Representatives—A. J. Hess, *Chairman*; Cecil Boling, E. H. Johnson, H. F. Spoehrer, F. L. Tarleton.

NRC-National Academy of Sciences, Div. of Engrg.: John W. James.

Building: P. J. Marschall, *Chairman*; Lester T. Avery, M. F. Blankin, C. R. Gardner, J. N. Livermore, R. A. Sherman.

Educational Committee: John W. James, *Chairman*; F. H. Buzzard, P. B. Gordon.

EJC Representatives: P. B. Gordon, A. J. Hess (Alternate), E. R. Queer (*Ex Officio*).

Exposition: L. N. Hunter, *Chairman*; P. B. Gordon, Walter A. Grant, J. W. James.

Joint ASHAE-ASAE Committee on Environment Studies: ASHAE Representatives—A. B. Algren, M. K. Fahnestock, A. J. Hess, H. A. Lockhart, H. E. Ziel.

AAAS Representative: Walter A. Grant.

Long-Range Planning: L. N. Hunter, *Chairman* (*one year*); John Everetts, Jr. (*one year*); J. W. James, J. D. Kroeker (*two years*); P. B. Gordon, John E. Haines (*three years*).

Nuclear Energy Engineering: W. F. Friend, *Chairman*; L. N. Hunter, R. A. Sherman, E. F. Snyder, Jr. (Committee on Research).

Publication Advisory: F. H. Faust, *Chairman*; R. E. Cherne, E. L. Crosby, J. C. Fitts, G. R. Munger, G. B. Priester.

U. S. National Committee of International Institute of Refrigeration: A. J. Hess, ASHAE Representative.

Nominating Committee

Region	*Member*	*First Alternate*	*Second Alternate*
1	W. C. Kruse, Jr.	E. K. Wagner	L. A. Childs
2	John Richmond	D. S. Falk	B. M. Kluge
3	D. M. Allen		G. W. F. Myers
4	J. E. Murray	L. A. O'Meara	R. M. Stern
5	R. O. McGary	R. K. Rouse	W. G. Robertson, Jr.
6	G. H. Meffert	J. S. Malahy, Jr.	F. R. Denham
7	D. W. Thomson		
2	A. W. Edwards*	N. H. Peterson* (4)	E. J. Morris* (5)
3	B. L. Evans*		
1	H. L. Martin*		
7	H. R. Roth*		

* Selected by Council; others selected by Chapters Regional Committees.

Local Chapter Officers—1958

Arizona

Organized 1953
Headquarters, Phoenix

President.......................F. W. Gabbard
Vice President................G. L. Jackson, Jr.
Secretary............................J. R. Hight
Treasurer.....................S. A. Fredrickson
Board of Governors: E. P. Boothroyd, S. A. Fredrickson, F. W. Gabbard, J. R. Hight, G. L. Jackson, Jr.

Arkansas

Organized 1952
Headquarters, Little Rock

President..........................K. A. Pettit
Vice President....................Allen Bullard
Secretary.....................O. L. McCallister
Treasurer.......................J. H. Lammons
Board of Governors: Fred Tenny, T. W. Turner

Atlanta

Organized 1937
Headquarters, Atlanta, Ga.

President............................J. S. Edgar
Vice President.......................W. P. West
Secretary..........................B. W. Dean
Treasurer.......................D. C. McNeil
Board of Governors: F. W. Bull, J. A. Marshall

Austin

Organized 1956
Headquarters, Austin, Tex.

President............................J. L. Rea
Vice President.......................J. R. Watt
Secretary.........................B. J. Barnhart
Treasurer.........................F. W. Gerling
Board of Governors: W. H. Luedecke, P. R. Sisler

Baltimore

Organized 1949
Headquarters, Baltimore, Md.

President.........................F. M. Hewitt
Vice President...................E. S. Berngartt
Secretary.....................C. R. Higdon, Jr.
Treasurer......................R. N. Flashman
Board of Governors: R. E. Dressell, J. F. Grice, R. F. Weisman

Baton Rouge

Organized 1955
Headquarters, Baton Rouge, La.

President.......................W. J. LeBlanc
Vice President.................A. J. Mayers, Jr.
Secretary.........................R. F. Dupuy
Treasurer.........................R. T. Collier
Board of Governors: D. W. Chesson, A. W. Holland, T. G. Neff

Bluegrass

Organized 1954
Headquarters, Louisville, Ky.

President...........................H. L. Carr
Vice President....................R. F. Logsdon
Secretary....................H. J. Bohannon, Jr.
Treasurer..........................S. S. Emison
Board of Governors: J. W. Frazier, O. J. Hill, R. W. Rademaker, C. J. Roberts

British Columbia

Organized 1952
Headquarters, Vancouver, B. C., Canada

President..........................S. W. Welsh
Vice President...................W. F. Wiggins
Secretary..........................D. B Leaney
Treasurer...........................R. W. Hole
Board of Governors: R. E. Atkey, R. A. Malcolm

Central New York

Organized 1944
Headquarters, Syracuse

President..........................E. L. Moyer
Vice President......................S. F. Gilman
Secretary...........................R. A. Barr
Treasurer..........................E. F. Smarzo
Board of Governors: R. A. Barr, E. L. Galson, S. F. Gilman, B. P. Morabito, E. L. Moyer, E. F. Smarzo, Merle Weninger

Central Ohio

Organized 1944
Headquarters, Columbus

President.........................R. C. Liebert
Vice President.......................J. W. Ford
Secretary.........................E. T. Stluka
Treasurer.........................R. L. Keener
Board of Governors: J. W. Ford, F. W. Gurley, R. L. Keener, W. R. Kinney, R. C. Liebert, E. T. Stluka

Chapitre de la Ville de Quebec

Organized 1956
Headquarters, Quebec, P.Q. Canada

President.....................Maurice Paquet
Vice President..................P. J. Lamarche
Secretary.....................Francois L'Anglais
Treasurer......................Azarias Servant
Board of Governors: Louis-Philippe Bonneau, O. P. Dorval, Jean-Marc Lagace, Lucien Larocque, Jean Veilleux

Cincinnati

Organized 1932
Headquarters, Cincinnati, Ohio

President.........................C. P. Krantz
Vice President.................E. A. Sobolewski
Secretary.........................W. H. Rieger
Treasurer.........................A. J. Staubitz
Board of Governors: C. J. Kummer, T. D. Reiley

Connecticut

Organized 1940
Headquarters, New Haven

President.......................... R. B. Cahoon
Vice President.................. F. J. Raible, Jr.
Secretary........................ E. M. Johnson
Treasurer............................ Eric Olsson
Board of Governors: F. Honerkamp, B. M. Packtor, W. J. Waeldner

Delta

Organized 1939
Headquarters, New Orleans, La.

President.......................... R. K. Goode
Vice President.................... O. F. Schully
Secretary........................ E. H. Sanford
Treasurer........................ J. H. Maloney
Board of Governors: W. B. Moses, Jr., R. D. Lewis

El Paso

Organized 1956
Headquarters, El Paso, Tex.

President...................... W. A. Byars, Jr.
Vice President........................ J. A. True
Secretary........................ J. H. Brooks
Treasurer............................ J. L. Huff
Board of Governors: M. D. Goodwin, J. W. Lantow

Empire State Capital

Organized 1951
Headquarters, Albany, N. Y.

President........................ E. J. Mahoney
1st Vice President............... M. E. Waddell
2nd Vice President................. B. E. Mullen
Secretary-Treasurer.............. E. E. Phillips
Board of Governors: F. R. Foote, J. L. Ottenheimer, W. A. Smith

Fort Worth

Organized 1957
Headquarters, Fort Worth, Tex.

President.................... T. B. Romine, Jr.
Vice President................ J. K. Mattox, Jr.
Secretary............................ C. L. Zahn
Treasurer.......................... G. F. Smith
Board of Governors: B. R. Olson, J. L. Tye

Golden Gate

Organized 1937
Headquarters, San Francisco, Calif.

President............................ J. B. Smith
Vice President.................. T. R. Simonson
Secretary.......................... R. C. Pribuss
Treasurer .... D. A. Delaney
Board of Governors: Herb Duncan, L. E. Dwyer, J. D. Kniveton

Illinois

Organized 1906
Headquarters, Chicago

President..................... Herbert Kreisman
Vice President.................. E. P. Heckel, Jr.
Secretary............................ J. C. Scott
Treasurer............................ L. H. Streb
Board of Governors: E. H. Gage, E. R. Gritschke, A. W. Lippitt

Illinois-Iowa

Organized 1956
Headquarters, Moline, Ill.

President.......................... D. H. Ninow
Vice President........................ E. O. Hull
Secretary........................ R. F. Demange
Treasurer........................ D. G. Johnson
Board of Governors: W. M. Curtin, M. L. Smith

Indiana

Organized 1943
Headquarters, Indianapolis

President.......................... W. F. Currise
Vice President.................... J. M. Teskoski
Secretary............................ A. B. Keller
Treasurer........................ W. L. Kercheval
Board of Governors: R. I. Drum, H. J. Kennedy, G. W. Vogel

Inland Empire

Organized 1950
Headquarters, Spokane, Wash.

President.......................... R. D. Nevers
Vice President........................ J. R. Morris
Secretary........................ R. B. Campbell
Treasurer.......................... J. L. Harvey
Board of Governors: J. A. Doyle, S. A. Thomas, K. M. Wood

Iowa

Organized 1940
Headquarters, Des Moines

President...................... R. T. Howard, Jr.
Vice President.................... C. B. Campbell
Secretary............................ J. R. Bain
Treasurer............................ J. L. Bouse
Board of Governors: L. R. Moore, E. O. Olson, N. L. Rutgers

Jacksonville, Florida

Organized 1957
Headquarters, Jacksonville

President.......................... J. R. Spence
Vice President...................... R. L. Burnes
Secretary.......................... W. D. Minick
Treasurer........................ W. P. Sherman
Board of Governors: H. A. Holborn, F. B. Wilder

Johnstown

Organized 1957
Headquarters, Johnstown, Pa.

President...................... K. O. Schlentner
Vice President.................... Sigmond Moroh
Secretary........................ J. K. Thornton
Treasurer.......................... L. W. Straw
Board of Governors: H. F. Lenz, W. L. Ross, L. F. Tierney

Kansas

Organized 1951
Headquarters, Wichita

President.......................... G. L. Oswald
Vice President.................. L. R. Martin, Jr.
Secretary.......................... R. B. Peugh
Treasurer.......................... D. L. Manson
Board of Governors: O. P. Bullock, T. L. Roberts, Charles Yoe

Local Chapter Officers—1958 (*Continued*)

Kansas City

Organized 1917
Headquarters, Kansas City, Mo.

President....................J. C. Fasnacht
Vice President....................L. A. Heaven
Secretary....................T. I. Harriman
Treasurer....................E. M. Hopkins
Board of Governors: A. S. Hurt, Jr., D. E. Huxtable, R. B. Luhnow, Jr., S. C. McCann, Gustav Nottberg

Long Island

Organized 1957
Headquarters, Garden City, L. I., N. Y.

President....................H. J. Campbell, Jr.
Vice President....................C. M. Alston
Secretary....................J. L. Page
Treasurer....................S. M. Walzer
Board of Governors: W. T. Cleland, W. H. Hoops, W. G. Kane

Manitoba

Organized 1935
Headquarters, Winnipeg, Man., Canada

President....................D. S. Swain
Vice President....................W. L. Algie
Secretary....................H. R. Skinner
Treasurer....................F. J. Lindenschmidt
Board of Governors: J. L. Greer, Stanley Hayden, K. J. McCartney

Massachusetts

Organized 1912
Headquarters, Boston

President....................R. F. Curry
Vice President....................F. J. Butler
Secretary....................R. B. Stevens
Treasurer....................R. E. Reid
Board of Governors: W. F. Lynch, D. W. Noble, G. T. Roberts, Jr.

Memphis

Organized 1944
Headquarters, Memphis, Tenn.

President....................G. B. Ramsey, III
Vice President....................H. W. Wade
Secretary....................A. L. Barksdale
Treasurer....................G. C. Crow, Jr.
Board of Governors: R. H. Bolding, W. L. Henson, John Hilton, III

Miami Valley

Organized 1950
Headquarters, Dayton, Ohio

President....................R. B. Walcott
Vice President....................N. O. Mitchell
Secretary....................W. H. Fogarty
Treasurer....................F. H. Doench, Jr.
Board of Governors: J. R. Ahart, L. P. Brehm, J. B. Rishel, R. E. Turlis, C. D. Weaver

Michigan

Organized 1916
Headquarters, Detroit

President....................K. J. Wagoner
Vice President....................J. B. Olivieri
Secretary....................J. G. Black, Jr.
Treasurer....................C. J. Henstock
Board of Governors: R. E. Maund, K. A. Nesbitt, R. E. Strand

Minnesota

Organized 1918
Headquarters, Minneapolis

President....................J. H. Jester
Vice President....................R. G. Gridley
Secretary....................J. W. McNamara
Treasurer....................C. T. Hastings
Board of Governors: G. F. Gausman, H. G. Sierk

Mississippi

Organized 1953
Headquarters, Jackson

President....................D. F. Ingram
Vice President....................F. H. North
Secretary....................W. D. Fortner
Treasurer....................G. H. Schmaltz
Board of Governors: M. E. Arledge, T. V. Curtis, T. D. Luke

Montreal

Organized 1936
Headquarters, Montreal, Que., Canada

President....................Arthur de Breyne
Vice President....................W. E. Jarvis
Secretary....................W. U. Robinson
Treasurer....................H. B. Cooper
Board of Governors: Robert Clapperton, Henri Dagenais, J. G. LeFrancois, H. G. S. Murray

Nebraska

Organized 1940
Headquarters, Omaha

President....................E. N. Seiler
Vice President....................F. P. Manchester, Jr.
Secretary....................W. L. Ryan
Treasurer....................C. L. Thomsen
Board of Governors: R. F. Cummings, G. C. Mittauer, M. F. Stober

New Mexico

Organized 1954
Headquarters, Albuquerque

President....................F. H. Bridgers
Vice President....................E. L. Brazier, Jr.
Secretary....................R. P. Lee
Treasurer....................V. J. Stephens
Board of Governors: W. M. Carroll, T. W. Englinton, H. K. Pride

New York

Organized 1911
Headquarters, New York

President....................W. J. Olvany
Vice President....................L. D. Carr
Secretary....................S. R. Apt
Treasurer....................S. A. Spencer
Board of Governors: P. A. Bourquin, W. T. Kane, Sherman Loud, C. M. Wilson

North Jersey

Organized 1952
Headquarters, Newark, N. J.

President....................S. H. Nitzberg
Vice President....................L. G. Huggins
Secretary....................C. E. Parmelee
Treasurer....................C. W. Zimmer
Board of Governors: Martin C. Christesen, G. Vernon Dennis, Herbert Fox, R. S. Taylor

Local Chapter Officers—1958 (*Continued*)

North Texas

Organized 1938
Headquarters, Dallas

President.......................... Ross Zumwalt
Vice President.................... J. D. Poythress
Secretary.......................... O. H. Mehl
Treasurer.......................... R. M. Kilpatrick
Board of Governors: C. F. Gilmore, W. S. Keeney, G. H. Meffert

Northeastern Oklahoma

Organized 1948
Headquarters, Tulsa

President.......................... J. C. Chase
Vice President.................... B. F. Purifoy
Secretary.......................... J. E. Tumilty
Treasurer.......................... W. C. Buckner
Board of Governors: C. H. Dollmeyer, R. F. Shoemaker, R. W. Winget

Northern Alberta

Organized 1956
Headquarters, Edmonton, Alta., Canada

President.......................... E. K. Cumming
Vice President.................... R. A. Williams
Secretary.......................... G. N. Campbell
Treasurer.......................... Kornelius Siemens
Board of Governors: Harry Hole, J. F. MacBride, J. S. Michener, R. G. Proudfoot, G. W. Sadler

Northern Ohio

Organized 1916
Headquarters, Cleveland

President.......................... R. A. Urban
Vice President.................... R. D. Wilson
Secretary.......................... B. A. Schwirtz
Treasurer.......................... R. M. Rubin
Board of Governors: J. L. Frissee, J. F. Koran, R. E. Leising

Northern Piedmont

Organized 1952
Headquarters, Greensboro, N. C.

President.......................... G. B. Rottman
Vice President.................... Boleslaw Jeglinski
Secretary.......................... E. A. Stroupe, Jr.
Treasurer.......................... M. W. Wasell
Board of Governors: C. Z. Adams, C. A. Cofer, A. H. Sprinkle, Jr.

Oklahoma

Organized 1935
Headquarters, Oklahoma City

President.......................... W. R. Johnson
Vice President.................... W. H. Stewart
Secretary.......................... T. L. Robinson
Treasurer.......................... S. R. Hill
Board of Governors: F. R. Denham, J. R. Patten, A. C. Shelley

Ontario

Organized 1922
Headquarters, Toronto, Ont., Canada

President.......................... Jack Thompson
Vice President.................... R. A. Ritchie
Secretary-Treasurer.............. H. R. Roth
Board of Governors: J. D. Coates, Ernest Fox, W. A. Mould, C. L. Torry

Oregon

Organized 1939
Headquarters, Portland

President.......................... Keith Kruchek
Vice President.................... F. T. Taylor
Secretary.......................... G. H. Lord, Jr.
Treasurer.......................... W. S. Cooper
Board of Governors: W. B. Hayes, C. W. Timmer, R. W. Zanders

Ottawa Valley

Organized 1952
Headquarters, Ottawa, Ont., Canada

President.......................... Jacob Klassen
Vice President.................... A. H. Hargreaves
Secretary.......................... C. N. Kirby
Treasurer.......................... G. A. Gray
Board of Governors: J. W. Green, G. W. Goodkey, I. M. Paterson, C. W. Watson

Philadelphia

Organized 1916
Headquarters, Philadelphia, Pa.

President.......................... C. J. Forve
1st Vice President................ J. J. Hucker
2nd Vice President............... Ludwig Mack
Secretary...... John Everetts, Jr., H. N. Teuber*
Treasurer.......................... D. S. Plewes
Board of Governors: P. R. Anderson, P. H. Yeomans

Pittsburgh

Organized 1919
Headquarters, Pittsburgh, Pa.

President.......................... E. J. Busch
Vice President.................... C. W. Stanger
Secretary.......................... G. E. Smetak
Treasurer.......................... R. C. Firsching
Board of Governors: G. W. Cost, A. F. Nass, Jr., E. H. Riesmeyer, Jr.

Puget Sound

Organized 1928
Headquarters, Seattle, Wash.

President.......................... M. W. McKinstry
1st Vice President................ H. M. Hendrickson
2nd Vice President............... R. H. Liniger
Secretary.......................... K. C. Massart
Treasurer.......................... D. M. Hopkins
Board of Governors: W. L. Johnson, A. T. Kane

Rocky Mountain

Organized 1944
Headquarters, Denver, Colo.

President.......................... J. R. Wilson
Vice President.................... R. T. Beck
Secretary.......................... R. J. Walker
Treasurer.......................... V. L. Wilkin
Board of Governors: Leo Krisl, E. A. Thompson

Sacramento Valley

Organized 1952
Headquarters, Sacramento, Calif.

President.......................... J. A. White
Vice President.................... R. T. Andrews
Secretary.......................... W. B. Lander
Treasurer.......................... E. J. Varvello
Board of Governors: M. J. Delavan, George McGrew, D. R.Peterson

---

* Filled unexpired term.

Local Chapter Officers—1958 (*Continued*)

St. Louis

Organized 1918
Headquarters, St. Louis, Mo.

President....................J. J. Blackmore
Vice President....................K. O. Williams
Secretary....................J. B. Killebrew
Treasurer....................H. R. Halt
Board of Governors: W. C. Bruce, N. J. Hubbuch, E. C. Kuntz, J. I. Levenhagen, C. J. McClure, J. S. Rosebrough

Savannah

Organized 1958
Headquarters, Savannah, Ga.

President............R. L. Reiley*, E. F. Young†
Vice President.....S. H. Baugh*, K. R. Goddard†
Secretary............R. E. Kinser*, R. L. Hume†
Treasurer............E. F. Young*, J. M. Cates†
Board of Governors: A. P. Gnann, Jr.*, K. R. Goddard*, C. M. Courtenay†, M. C. Nettles†

Shreveport

Organized 1948
Headquarters, Shreveport, La.

President....................L. E. Kneipp
Vice President....................J. L. Collins, Jr.
Secretary....................J. J. Guth, Jr.
Treasurer....................R. L. Johnson, Jr.
Board of Governors: A. L. Jones, Jr., F. H. Spaulding, Jr., J. S. Tarlton

South Carolina

Organized 1954
Headquarters, Columbia

President....................H. J. Haar, Jr.
Vice President....................J. D. Williams
Secretary....................T. O. Curlee, Jr.
Treasurer....................W. O. Blackstone
Board of Governors: F. A. Bailey, III, M. R. Durlach, Jr., B. A. Leppard

South Texas

Organized 1938
Headquarters, Houston

President.......A. B. Ullrich, Jr.‡, E. B. Appling¶
Vice President.E. B. Appling‡, H. D. McMillan, Jr.¶
Secretary....H. D. McMillan, Jr.‡, H. G. McKee¶
Treasurer.........H. G. McKee‡, E. H. McLane¶
Board of Governors: E. H. McLane‡, P. C. Nail‡, J. M. Daniel¶, P. C. Nail¶

Southern Alberta

Organized 1958
Headquarters, Calgary, Alta., Canada

President....................H. W. Klassen
Vice President....................L. D. Ontonio
Secretary....................P. M. Meis
Treasurer....................W. R. Laing
Board of Governors: E. W. Deeves, N. J. Howes, F. B. Thompson, E. H. Watson

---

* These officers were elected for year 1957–58, but were not recorded in 1957 TRANSACTIONS, because chapter had not yet been organized.
† These officers are for the year 1958–59.
‡ Elected October 18, 1957
¶ Elected January 7, 1958 to fill unexpired term

Southern California

Organized 1930
Headquarters, Los Angeles

President....................A. Z. Levine
Vice President....................R. C. Taylor
Secretary....................Albert Zimmerman
Treasurer....................B. L. Hutchinson, Jr.
Board of Governors: J. M. Ayres, C. J. Bressoud, Will Geselbracht, Jack Miller

Southern Piedmont

Organized 1952
Headquarters, Charlotte, N. C.

President....................J. R. Clark
Vice President....................N. W. McGuire, Jr.
Secretary.....Wayne Bainbridge, E. V. Overcash*
Treasurer......W. P. Wells, Jr., W. P. Smith, Jr.*
Board of Governors: E. H. Farthing, G. C. Garrett, H. A. Wright

Southwest Texas

Organized 1946
Headquarters, San Antonio

President....................J. G. Ford
Vice President....................K. A. J. Monier
Secretary....................E. E. Cravens
Treasurer....................C. J. Troilo
Board of Governors: G. M. Baker, C. L. Herndon, M. E. Staley

Toledo

Organized 1954
Headquarters, Toledo, Ohio

President....................D. L. Wilson
Vice President....................G. R. Munger
Secretary....................E. J. Katafiasz
Treasurer....................N. W. Dawe
Board of Governors: F. A. Edgington, T. F. Geikie, Robert Greenwald, J. E. Wilkie

Utah

Organized 1944
Headquarters, Salt Lake City

President....................L. K. Irvine
Vice President....................R. V. Oliver
Secretary-Treasurer..............V. Q. Tregeagle
Board of Governors: R. C. Evans, C. E. Ferguson, Fred Richeda, D. R. Wilde

Virginia

Organized 1946
Headquarters, Norfolk

President....................D. L. Gusler
Vice President....................J. R. Spencer
Secretary....................R. F. Fox
Treasurer....................A. R. Thompson, Jr.
Board of Governors: J. F. Boyenton, D. C. Delinger

---

* Filled unexpired term.

Local Chapter Officers—1958 (*Continued*)

Washington, D. C.

Organized 1935
Headquarters, Washington, D. C.

President.........................J. W. Morgan
Vice President....................H. E. Grossman
Secretary.........................L. D. Pain
Treasurer.........................R. J. Ruschell
Board of Governors: R. F. Dovenor, H. W. Rush, H. G. Strong

West Texas

Organized 1953
Headquarters, Lubbock

President.........................W. R. Anthony
Vice President....................C. P. Houston
Secretary.........................R. L. Mason
Treasurer.........................O. R. Downing
Board of Governors: M. J. Aderton, J. F. Roberts

Western Massachusetts

Organized 1955
Headquarters, Springfield

President.........................C. R. Munn
Vice President....................J. J. Curran
Secretary.........................K. W. Maki
Treasurer.........................T. E. Fallon
Board of Governors: R. E. Cross, F. A. Ferraro, R. J. Hildreth, A. M. Lovenberg, J. E. Reed

Western Michigan

Organized 1931
Headquarters, Grand Rapids

President.........................B. J. Waalkes
Vice President....................S. R. Curtis
Secretary.........................G. L. Jepson
Treasurer.........................R. B. Smith
Board of Governors: R. E. Distel, P. S. Morton, B. J. Walter

Western New York

Organized 1919
Headquarters, Buffalo

President.........................J. P. Guerra
1st Vice President................J. F. Bedard
2nd Vice President................D. J. Seifert
Secretary.........................F. R. Collins, Jr.
Treasurer.........................M. C. Beman
Board of Governors: Joseph Davis, Roswell Farnham, G. E. Kuhn, C. W. Stone, Q. P. Thompson

Wisconsin

Organized 1922
Headquarters, Milwaukee

President.........................R. D. Rodwell
Vice President....................A. A. Stutheit
Secretary.........................R. I. Anderson
Treasurer.........................J. E. Illingworth
Board of Governors: H. W. Alyea, H. D. Cook, I. J. Rossiter

Overseas Branch

Switzerland

Organized 1952
Headquarters, Zurich

President.........................A. Eigenmann
Vice President....................R. Goerg
Secretary.........................H. Kamm
Treasurer.........................W. Niederer
Headquarter.......................Dr. H. Brown

Student Branches

North Carolina State College

Organized 1948
Headquarters, Raleigh

Faculty Adviser..............Prof. R. B. Knight

Oregon State College

Organized 1949
Headquarters, Corvallis

Faculty Adviser..........Prof. G. E. Thornburgh

Purdue University

Organized 1949
Headquarters, W. Lafayette, Ind.

Faculty Adviser...............Prof. F. B. Morse

Texas A & M College

Organized 1946
Headquarters, College Station

Faculty Adviser............Prof. L. S. O'Bannon

University of Detroit

Organized 1949
Headquarters, Detroit, Mich.

Faculty Adviser....................J. B. Olivieri

University of Toronto

Organized 1951
Headquarters, Toronto, Ont., Canada

Faculty Adviser...............Prof. F. G. Ewens

TRANSACTIONS

of

American Society of Heating and Air-Conditioning Engineers

No. 1620

SIXTY-FOURTH ANNUAL MEETING, 1958

Pittsburgh, Pennsylvania

DURING the seven sessions of the 64th Annual Meeting January 27–29, 1958, action was taken on important business items. These included the reports of the Officers and of the Council; a report on election of Officers; results of a vote on various proposed By-Law amendments and on an amendment to the employees retirement plan; appropriate resolutions concerning the meeting were adopted; and a press release on the merger plan with ASRE was issued. The newly elected Officers and Council members were installed in office.

Of the seven sessions on the program (see page 24) at the Annual Meeting, two were of the symposium type and one was a topical session. A total of 17 technical papers were presented of which one was made available by title only. All these papers have been printed in the Journal. In addition, 10 papers were prepared for the 2 symposium sessions.

While attendance records for annual meetings were not broken, still available facilities were taken by the total of 945 who registered, including 596 members, 208 guests and 146 ladies, coming from 38 states and Canada with one member present from England. With most of the programmed events taking place on a single floor at the Penn-Sheraton Hotel, the Committee on Arrangements of the Pittsburgh Chapter succeeded in holding attention and handling all events very well.

Supplementing the technical sessions, the Committee on Arrangements of the Pittsburgh Chapter prepared the program to take care of the interests of all in

attendance. This included a series of events for the ladies with a tea and entertainment, and a bus tour of the city. For everyone, there were inspection trips on 2 afternoons.

At the Opening Luncheon on Monday, certificates were awarded to the newly-elected Fellows: A. L. Baum, New York, N. Y.; A. C. Buensod, New York, N. Y.; C. F. Kayan, New York, N. Y.; I. W. Cotton, Indianapolis, Ind.; E. L. Crosby, Baltimore, Md.; E. P. Heckel, Chicago, Ill.; F. B. Rowley, Minneapolis, Minn.; and G. M. Simonson, San Francisco, Calif.

In accordance with custom, the Annual Banquet on Wednesday evening included not only an address by Allison R. Maxwell, Jr., Pittsburgh, Pa., but also was the occasion for the installation of the new officers and the presentation of the F. Paul Anderson Medal and the Past President's Award. Thomas Pratt, Pittsburgh architect, acted as toastmaster, with the Invocation being given by Dean N. R. H. Moor, D.D., Trinity Episcopal Cathedral, Pittsburgh.

Each year, the Annual Meeting permits a review of the Society Research Program and activities and much advance planning is carried on. This is done through meetings of the Committee on Research and its Technical Advisory Committees.

The Executive Committee of the Committee on Research met on Saturday and this was followed by the meeting of the entire Committee on Sunday. On both Monday and Tuesday afternoons and Tuesday evening, meetings of the various Technical Advisory Committees were in session.

Technical Advisory Committees which held meetings included Air Cleaning, Heat Transfer through Fenestration, Hot Water and Steam Heating, Sorption, Heat Pump, Combustion, Industrial Environment, Solar Energy Utilization, Weather Data, Physiological Research, Evaporative Cooling, Heating and Air Conditioning Loads, Insulation, Odors, and Sound and Vibration Control.

Honoring of outstanding members is one of the features of each Annual Meeting. Two forms of honor established by gifts or endowments are the F. Paul Anderson Award consisting of a gold medal and citation, and the ASHAE-Homer Addams Award consisting of a scholarship for graduate work and a certificate.

Each year too, the retiring president receives the Past President's Award from his predecessor in that office.

The award of the F. Paul Anderson Medal was the 16th made since its establishment in 1930 and went to Prof. G. L. Tuve, Case Institute of Technology, Cleveland, Ohio—a past president of the Society, with the citation reading: Engineer, educator, researcher and author, who has performed noteworthy services for the engineering profession and for the benefit of the general public by training men in heating, ventilating, cooling and air conditioning and through his dedication to research in these arts and sciences.

Following the presentation of the medal at the Annual Banquet, Professor Tuve responded with the following remarks:

This honor certainly has been a surprise. I can make no adequate response. I can only say a humble and sincere "Thank You". There is no greater honor for a professional man than that of being recognized by his colleagues. I don't deserve a place among the illustrious recipients of the F. Paul Anderson Medal.

May I add also another word of thanks. This on behalf of my colleagues who are faculty members in the engineering colleges, and also members of this Society. We are proud to take part in the technical activities of the ASHAE. We value our share in its friendships. Dean F. Paul Anderson was one of us, and there are good and loyal members of ASHAE on every major engineering faculty in the United States and Canada. Very few engineering societies can say this. The unusual part is the way this Society

gets the professors to work. But there is good reason for this active work. No other Society has such a research program. No other society keeps its members so technically up-to-date by a yearly GUIDE or handbook. We in the universities are grateful for the opportunity of working with our fellow engineers in business and industry on such forward-looking technical programs. I hope that your committees on Long Range Planning and Inter-Society Merger will protect these valuable assets. I am certain that our great program of technical committee assignments is a major reason for the enthusiasm of our members.

A member of a busy technical committee may speak of *contributing* a lot of time. But each of us is very highly paid. The associations and friendships with our fellow members are high personal rewards.

I first worked on Society Research under our first Director, Dean John H. Allen, and under Professor Rowley, while I was in college. I was privileged to know F. Paul Anderson, and also Ferry Houghten and Cyril Tasker, and several generations of Society officers. I have seen dozens of Society Research projects in our colleges, and I know that these pay off for both parties. So I wanted to tell you how much we in the colleges value our membership in ASHAE and our friendships with you members in industry. Thank you again.

The certificate accompanying the second ASHAE-Homer Addams Award was presented at the Welcome Luncheon by Paul K. Addams, New York, N. Y., and was received by W. E. Springer, a graduate student at the University of Illinois.

The Past President's Award was made by Past Pres. John W. James, Chicago, Ill., to retiring Pres. P. B. Gordon, New York, N. Y.

At their luncheon meeting, life members decided to award a plaque to Prof. Linn Helander to convey their regard for his work in this field. This award was made at a meeting of the Kansas City Chapter on February 3.

A conference was also held on Tuesday evening for the benefit of the editors of chapter publications. It attracted attendance from a considerable fraction of these editors. Its purpose is to serve as a meeting place where their particular problems can be discussed.

The Council of the Society met on Sunday preceding the Meeting and the organization of the 1958 Council took place at a meeting held on Thursday immediately after the close of the Meeting.

FIRST SESSION, MONDAY, JANUARY 27, 9:00 A.M.

After calling the session to order, Pres. P. B. Gordon, New York, N. Y., called on T. F. Rockwell, Chairman of the Committee on Arrangements of the Pittsburgh Chapter, and he welcomed the members to Pittsburgh and outlined the chief features of the program as prepared by the committee.

REPORT OF PRESIDENT

President Gordon then presented his annual report in which he stated that during the year there had been an increase in membership of nearly 700, but that there were indications that the recent growth pattern may be changing, that the Regional Plan has passed through its second year, that the problems of Long Range Planning are tied to both rate of growth and financing, that operating costs may soon need corrective measures, that a professional investment advisory service had been retained, that a new Director of Research had taken office during the year, that improvements in technical information had been made available, and that a new Education Committee had been authorized. He also reported on Council actions

concerning the proposed merger of ASHAE and ASRE (the full text of the Report of the President appears on pages 173 and 174 of ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March 1958).

REPORT OF THE COUNCIL

Presented here is the 1957 Annual Report of the Council of the Society, as required by the Membership Corporations Law of the State of New York. Included also is a summary of the Treasurer's Report, a part of which is the Accountant's Report (see pages 7 to 12).

Four quarterly meetings of the Council were held during the calendar year 1957, the organization meeting being convened at the Conrad Hilton Hotel, Chicago, Ill., February 28; the spring meeting on May 4 at The Roosevelt Hotel, New Orleans, La.; the summer meeting on June 23 at the Manoir Richelieu, Murray Bay, Quebec, Canada and the fall meeting, October 12 at the Hotel Statler, Cleveland, Ohio.

At the organization meeting the Council approved the personnel of the Council, General and Special Committees and appointed the following Regional Directors: John Everetts, Jr., Region 1; A. B. Algren, Region 3; C. B. Gamble, Region 5 and D. M. Mills, Region 6.

A procedure for the election of Honorary Members and the conferring of the grade of Fellow was adopted; renewal of the ASHAE-*ASRE* Exchange Service Plan was authorized and depositories for Society funds were approved.

Among other important actions taken by the Council were the following:

1. Approved 5 charters for Chapters at (*a*) Johnstown, Pa.; (*b*) Jacksonville, Fla.; (*c*) Long Island, N. Y. (Garden City); (*d*) Savannah, Ga., and (*e*) Southern Alberta (Calgary).
2. Appointed B. H. Jennings as the new director of research effective August 1, and F. W. Hofmann as assistant secretary.
3. Elected 1 Honorary Member, 23 Life Members and conferred the grade of Fellow on 8 members.
4. Announced the personnel of the Guide Committee (9 members) with H. W. Alyea as Chairman and selected 5 nominees for a three-year term on the Committee on Research.
5. Authorized the revision of the employees' pension plan and a new group insurance plan.
6. Selected Prof. G. L. Tuve as the recipient of the F. Paul Anderson Medal and requested a nomination from the University of Illinois for the ASHAE-Homer Addams Award.
7. Recommended By-Law revisions to enlarge Council Committees and change membership qualifications. It also approved Chapter By-Law changes for Chapters, a new Operational Guide for the Nominating Committee and revisions for the Operational Guide of the Committee on Research.
8. New rules were adopted on the publication of Symposium talks.
9. A new Advisory Investment Service was authorized; a new rate schedule for the 1959 Guide was approved and some additional rules on travel expenses were adopted. Also, the Council adopted a budget of $706,995.00 for the fiscal period, November 1, 1957 to October 31, 1958.
10. Dates were selected for the Semi-Annual Meetings in 1959 and 1960, the 67th Annual Meeting in 1961 and the 69th Annual Meeting in 1963, as well as a schedule for Chapters Regional Committee meetings in 1958.

11. Council decided to establish Society headquarters in the proposed Engineering Societies Center in New York when the new facility is opened in 1960.

12. As required by Section 46 of the Membership Corporations Law of the State of New York, the following financial report is presented by the Council and filed herewith:

Assets—October 31, 1957

The Society has the following Assets:

Land and Laboratory Buildings at Cleveland, Ohio	$ 81,012.89
Furniture and Equipment, Tools, etc. at New York and Cleveland	37,669.30
On deposit in New York and Cleveland banks	291,671.81
Securities and Accrued Interest (Investment Account) with Bankers Trust Co., New York	181,364.69
Accounts Receivable	24,097.20
Inventories	6,211.34
Prepaid Expenses	11,956.49
TOTAL ASSETS—1957	$633,983.72
TOTAL ASSETS—1956	500,891.00
Increase for the year	$133,092.72

Liabilities—October 31, 1957

The Society has the following Liabilities:

Accounts Payable	$ 885.54
Federal Withholding Taxes	1,956.00
Accrued Accounts	15,550.00
Deferred Income:	
Research Projects	2,321.51
Prepaid Dues, Admission Fees, etc.	30,184.46
Reserve for	
Fluctuation in Canadian Exchange	122.71
TOTAL LIABILITIES	$ 51,020.22
NET WORTH (FUNDS)	582,963.50
TOTAL LIABILITIES AND NET WORTH	$633,983.72

At the end of the fiscal year, October 31, 1957, the Balance Sheet shows the Society's net worth as $582,963.50 compared to $459,751.92 on October 31, 1956, a net gain of $123,211.58.

Total cash in New York and Cleveland banks $291,671.81 consisting of $37,467.04 in General Operating Fund, $20,709.61 in Reserve Fund, $366.45 in F. Paul Anderson Fund, $57,757.55 in Research General Fund, $163,233.31 in Research Reserve, $807.27 in Research Endowment, $8,204.55 in Building Maintenance Reserve Fund, $449.35 in Building Fund, $294.19 in the ASHAE-Homer Addams Fund, $51.74 in the Education Fund of Life Members, $2,046.49 in the Gustus L. Larson Fund and $284.26 in the Investment Account at Bankers Trust Co., New York.

Now in the Investment Account at the Bankers Trust Co., New York, there are securities as follows: General Fund $13,178.92; Building Fund $38,814.60; Reserve Fund $119,158.01; F. Paul Anderson Award Fund $954.31, and Homer Addams Fund $9,543.11.

The Society's Reserve Fund now totals $139,867.62. As of October 31, 1957 the Research Reserve Fund had a balance of $163,233.31.

The Sources of Society Income were: Admission Fees and Dues $174,082.92; Publications $231,811.82; Interest $621.55; Research $338,067.55; Other Income $1,115.67—Total $745,699.51.

Society Expenditures were: Meetings, Committees and Chapters $43,549.96; Publications $141,475.88; Headquarters $233,192.68; Research $207,946.56—Total $626,165.08.

The insurance coverage of the Society is as follows: Fire for Buildings $201,000.00; for Contents $147,000.00; Sprinkler $15,000.00; Automobile Comprehensive Public Liability $100,000.00–300,000.00; Property Damage $25,000.00; Medical Payments $100,000.00; Bodily Injury $25,000.00–100,000.00; Public Liability, Premises, New York and Cleveland $100,000–300,000.00; Fidelity Bonds on Officers and all Employees $20,000.00. Workmen's Compensation is carried in New York and Ohio as required by law.

The Society has contributed to the Employees' Retirement Plan Trust $3,961.99 and the participating employees contributed a substantially similar amount deducted from salary.

The Society admitted the following members in the grades indicated since January 1, 1957:

Members	348
Associate Members over 30	370
Associate Members under 30	141
Affiliates	417
Students	152
Total	1,428

During the period January 1, 1957 to December 31, 1957, a total of 62 members in the Member, Associate Member, Affiliate and Student grades were reinstated; a total of 736 in all categories were dropped from the rolls: 196 resigned, 491 were cancelled and death claimed 49.

Net gain of members was 692. The overall status from January 1, 1957 as compared with January 1, 1958 is as follows:

	January 1, 1957	January 1, 1958
Honorary	4	5
Fellows	5	12
Presidential Members	22	20
Life Members	227	240
Members	4,653	4,886
Associate Members (over 30)	4,080	3,646
Associate Members (under 30)	—	594
Affiliates	1,848	2,130
Students	182	180
	11,021	11,713

The names and addresses of the candidates for membership were published in the JOURNAL of the Society each month during the year and are on record in the Secretary's office. The present membership total is 11,713.

Respectfully submitted,

P. B. GORDON, *President*

C. H. PESTERFIELD, *Treasurer*

Accountant's Report

FRANK G. TUSA & CO.

CERTIFIED PUBLIC ACCOUNTANTS

37 Wall St., New York 5, N. Y.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.
62 WORTH ST., NEW YORK, N. Y.

Gentlemen:

In accordance with the authority contained in the organizational meeting of the 1957 Council on February 28, 1957, we have examined the books of account of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.—New York and Cleveland for the fiscal year ended October 31, 1957.

Our examination consisted of the verification of Assets and Liabilities as of the close of business October 31, 1957, and as sufficient a review of the income and expense accounts as we considered necessary to reflect, properly, the operations for the year then ended. This examination was conducted in accordance with generally accepted auditing standards, except as otherwise noted in the comments section, and included such auditing procedures and such other tests of the accounting records as we considered necessary under the circumstances.

We are pleased to present our report of this examination which includes the various statements and schedules listed in the table of contents. In addition thereto,we should like to direct your attention to the following comments which constitute an integral part of this report and which define the scope of our examination.

ASSETS

CASH

On Hand	$ 150.00	
Investment Advisory Account	284.26	
In Checking Accounts	76,992.24	
In Savings Banks	214,245.31	$291,671.81

All cash was verified either by count or by direct communication with the depositories. All balances reported to us were reconciled to those reflected on the books of account and were correct. Every depository responded.

DUES AND ACCOUNTS RECEIVABLE

The following classification of Dues and Accounts Receivable were submitted to us by the internal accounting staff:

MEMBERSHIP		
Members	$ 5,842.41	
Associate Members (Over 30)	8,140.88	
Affiliates	5,621.25	
Associate Members (Under 30)	935.25	
Students	11.00	$ 20,550.79
SUNDRY DEBTORS		9,377.03
		$ 29,927.82

We reviewed the system of internal control used in the accounting for the above receivable and found it to be adequate. No outside confirmation was requested of accounts receivable. We were, however, by means of other auditing procedures able to confirm substantially the above balances which in our opinion are correct. In the determination of the adequacy of the allowances for dues and accounts doubtful of collection we have reviewed the budgetary provisions for the fiscal year ended October 31, 1957 and such other pertinent data as we considered necessary. It is our opinion that the allowances for these doubtful accounts will be ample in absorbing possible losses due to non-collection for the budgeted year 1958.

The Research General Fund Accounts Receivable consists of a billing charge to the U. S. Navy Department for work completed during October, 1957 in the amount of $409.93, and a balance of $19,733.63 due from the Society General Fund for 40% of the dues specifically allocated to Research. The counterpart of this latter receivable of $19,733.63 is reflected as a current liability of the Society General Fund on page *10**. This $19,733.63 includes $7,841.82 of Dues Receivable which will be paid to Research when collected by the Society.

INVESTMENTS

A schedule of investments is to be found on page *15* of this report. In accordance with the authority contained in the Council meetings of February 24, May 3, and June 23, 1957 and the directions contained therein, the Bankers Trust Company, New York was retained to provide a continuing advisory service for the investments of Society cash as allocated by the Council upon recommendation by the Finance Committee in cooperation with its Investment Sub-Committee.

Confirmation was received from the Bankers Trust Company of the investment portfolio held for the credit of the Society as at October 31, 1957. A statement of the Investment Advisory Account is to be found on page *14* of this report. This statement reflects the individual undivided interests of each participating fund after they have been charged with their proportionate share of losses resulting from a recognition of the decline of $9,238.90 in the market value of the investment portfolio.

INVENTORIES

The following inventories were verified either by physical count or by direct communication with the printers.

TRANSACTIONS	$ 5,372.88
Emblems	538.46
GUIDE Paper	300.00
	$ 6,211.34

A schedule of TRANSACTIONS inventories follows:

Volume	Year	Quantity	Cost of each	Total cost
1–56	1895–1950	5,203	0.4000	$2,081.20
57	1951	640	2.2750	1,456.00
58	1952	123	1.7960	220.91
59	1953	408	2.000	816.00
60	1954	145	2.0665	299.64
61	1955	135	2.0742	280.02
62	1956	103	2.1273	219.11
				$5,372.88

ADVANCE DEPOSIT—AIR TRAVEL $ 425.00

This deposit placed with the United Airlines was confirmed by direct communication.

PROPERTY FUND ASSETS (Page 12) $118,682.19

The Land and Buildings, Instruments, Equipment and Furniture and Fixtures are reflected on the Statement of Financial Condition at cost of acquisition. With the exception of Land and Buildings all Assets have been depreciated at the rate of 10% per annum.

In accordance with the resolution adopted by the Council at its meeting of January 23, 1949, depreciation was not provided on the buildings for the current fiscal year.

During the current year, in response to a request of the Building Committee, the realty located at 7218 Euclid Ave., Cleveland, Ohio, was appraised by Mr. Ben B. Byer of Cleveland, Ohio, who submitted the following *Market Values* under report dated October 4, 1957:

Land	$ 50,000.00
Buildings	75,000.00
TOTAL APPRAISED MARKET VALUE	$125,000.00

* Page numbers refer to table of contents in accountant's submitted report.

Deferred Income	$ 25,869.14

The membership classfication of the dues prepaid by elected members follows:

Membership	
Members	$ 12,420.26
Associate Members (over 30 years)	7,575.38
Affiliates	4,619.50
Associate Members (under 30 years)	1,184.00
Students	45.00
Fellows	25.00
	$ 25,869.14

Funds

There is included as a part of this report, a Statement of Funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1957.

Insurance

The insurance coverage of the Society follows:

Fire		
Building (Cleveland)		$201,000.00
Personal Property:		
Cleveland	$104,000.00	
Printers	17,000.00	
New York	26,0 0.00	147,000.00
Sprinkler Leakage		
Waverly Press	10,000.00	
Horn-Shafer Co.	5,000.00	15,000.00
Non-Ownership—Automobile—Comprehensive		
Public Liability		100/300M
Property Damage		25M
Medical		1M
General Public Liability—Premises		
New York and Cleveland		
Liability		100/300M
Bodily Injury		25/100M
Commercial Blanket Bond		
President, Treasurer, Chairman of the Finance Committee, and First and Second Vice-Presidents		20,000.00

Operations

For the fiscal year ended October 31, 1957, the operations of the Society resulted in an Income in Excess of Expenses, in the amount of $118,864.14 as compared to $33,031.89 in the prior year or an increase of $85,832.25. However, it should be noted that the current years income included Exposition Contribution of approximately $132,000.00. Without this income, Society operations would have been shown an Excess of *Expenses over Income* of approximately $46,000.00 for the current year, or an unfavorable change of approximately $79,000,00 as compared to the prior year. This unfavorable change is principally due to an increase of almost $60,000.00 in Headquarters Expenses. Of this increase, approximately $14,000.00 represents a charge for Dues and Accounts Doubtful of collection—Salary and wage cost have increased by $36,600.00—the balance of $9,400.00 being reflected in the general rise of administrative costs of rent, telephone and telegraph, printing and stationery and others. A condensed comparative Statement of Income and Expense is to be found on page *24* of this report.

CONCLUSION

In our opinion the accompanying Statement of Financial Condition, the related Statement of Income and Expense and Statement of Fund Balances present fairly the financial condition of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., New York, N. Y. at October 31, 1957 and the results of its operations for the year then ended in conformity with generally accepted accounting principles applied on a basis consistent with that of the preceding year.

Dated December 11, 1957 FRANK G. TUSA & CO., CERTIFIED PUBLIC ACCOUNTANTS

STATEMENT OF FINANCIAL CONDITION—October 31, 1957

ASSETS

GENERAL FUNDS				
Society—New York			$ 90,532.25	
Research—Cleveland			79,869.92	$170,402.17
RESERVE FUNDS				
Society—New York				
Cash on Deposit		$ 20,709.61		
Investments				
Undivided interest in Investment Advisory Account	$125,218.55			
Less: Allowance for Decline in Market Value thereon	6,060.54	119,158.01	139,867.62	
Research—Cleveland				
Cash on Deposit			163,233.31	303,100.93
CAPITAL ASSET FUNDS				
Property Fund			118,682.19	
Building Fund				
Cash on Deposit		449.35		
Investments				
Undivided interest in Investment Advisory Account	40,788.75			
Less: Allowance for Decline in Market Value thereon	1,974.15	38,814.60	39,263.95	
Building Maintenance Reserve Fund				
Cash on Deposit			8,204.55	166,150.69
ENDOWMENT FUNDS				
F. Paul Anderson Award				
Cash on Deposit		366.45		
Undivided interest in Investment Advisory Account	1,002.85			
Less: Allowance for Decline in Market Value thereon	48.54	954.31	1,320.76	
Homer Addams				
Cash on Deposit		294.19		
Undivided interest in Investment Advisory Account	10,028.49			
Less: Market Decline	485.38	9,543.11	9,837.30	
Educational Fund—Life Members			51.74	
Gustus L. Larson Fund			2,046.49	
Research Endowment Fund			807.27	14,063.56
				$653,717.35

LIABILITIES AND NET WORTH

LIABILITIES

GENERAL FUND				
Society				
Federal Withholding Taxes		$ 1,956.00		
Due to Research General Fund (Per Contra)		19,733.63		
Sundry Debtors' Credit Balances		668.15		
Accrued Accounts				
Salaries	$ 13,800.00			
Professional Fees	1,750.00			
Junior Student Member Award	100.00	15,650.00		
Prepaid Dues and Fees				
Candidates' Dues	1,436.00			
Admissions Fees	2,879.32			
Cash on Deposit (Per Contra)		4,315.32		
Deferred Income				
Elected Members' Dues		16,023.09		
Reserve for Fluctuation-Canadian Exchange		122.71	$ 58,468.90	
Research				
Sundry Accounts Payable		117.39		
Deferred Income				
Elected Members Dues	9,846.05			
Unexpended Earmarked Contributions	2,321.51	12,167.56	12,284.95	
TOTAL LIABILITIES				$ 70,753.85

NET WORTH

GENERAL FUNDS			
Society—New York	32,063.35		
Research—Cleveland	67,584.97	99,648.32	
RESERVE FUNDS			
Society—New York	139,867.62		
Research—Cleveland	163,233.31	303,100.93	
CAPITAL ASSET FUNDS			
Property Fund	118,682.19		
Building Fund	39,263.95		
Building Maintenance Reserve Fund	8,204.55	166,150.69	
ENDOWMENT FUNDS			
F. Paul Anderson Award	1,320.76		
Homer Addams	9,837.30		
Gustus L. Larson	2,046.49		
Educational Fund—Life Members	51.74		
Research Endowment	807.27	14,063.56	
TOTAL NET WORTH			582,963.50
			$653,717.35

CONDENSED COMPARATIVE STATEMENT OF INCOME AND EXPENSES

INCOME	OCTOBER 31, 1957	OCTOBER 31, 1956	INCREASE (DECREASE)
SOCIETY			
Dues	$1[illegible]1,353.67	$148,820.13	$ 12,533.54
Admission Fees	11,555.00	9,741.00	1,814.00
Emblems, Pins, Etc.	1,174.25	961.80	212.45
Publications			
JOURNAL Contract	30,000.00	28,961.70	1,083.30
THE GUIDE	186,072.14	187,744.02	(1,671.88)
TRANSACTIONS	2,291.65	1,704.20	587.45
Books, Reprints, Codes, Etc.	7,437.98	5,314.72	2,123.26
Membership Roll	6,010.05	3,945.96	2,064.09
Exposition Contribution	132,002.13	-0-	132,002.13
Investments	621.55	664.27	(42.72)
Foreign Exchange	488.34	65.31	423.03
Miscellaneous	627.33	169.90	457.43
RESEARCH			
Dues	99,967.11	79,491.75	20,475.36
Contributions—General	21,913.60	22,079.63	(166.03)
Contributions—Earmarked	80,562.05	106,217.82	(25,655.77)
Interest	900.94	481.35	419.59
U. S. Navy—Bu. Ships	2,721.72	127.50	2,594.22
TOTAL INCOME	745,699.51	596,446.06	149,253.45
EXPENSES			
COMMITTEES AND CHAPTERS	32,120.74	24,595.87	7,524.87
MEETINGS	11,429.22	6,656.99	4,772.23
PUBLICATIONS			
Members' Subscriptions to H.P.A.C.	526.36	3,771.25	(3,244.89)
TRANSACTIONS	26,957.80	27,785.81	(828.01)
Membership Roll, Books, Reprints, Codes, Etc.	13,635.60	6,164.53	7,471.07
THE GUIDE	100,356.12	102,767.75	(2,411.63)
HEADQUARTERS	233,192.68	173,484.59	59,708.09
RESEARCH			
Promotion—Fund Raising	16,693.67	19,375.65	(2,681.98)
Committee Expenses	3,739.79	3,435.14	304.65
Staff Salaries	107,766.64	107,862.52	(95.88)
Laboratory	27,464.49	25,598.10	1,866.39
Building Operation and Maintenance	11,620.72	11,915.97	(295.25)
Provision for Building Maintenance Reserve	2,000.00	2,000.00	-0-
Cooperative Research	38,661.25	48,000.00	(9,338.75)
	626,165.08	563,414.17	62,750.91
NET OPERATING REVENUES	119,534.43	33,031.89	86,502.54
OTHER DEDUCTIONS			
Provision for Decline in Market Value of Investments	670.29	-0-	670.29
NET EXCESS OF INCOME OVER EXPENSES	$118,864.14	$ 33,031.89	$ 85,832.25

President Gordon then recognized Walter A. Grant, Syracuse, N. Y., chairman of the Finance Committee, who proposed the following:

RESOLVED that the Society hereby ratifies, confirms and approves in all respects the action of the Council in adopting said First Amendment to the Society's said Employees' Retirement Plan, Agreement and Declaration of Trust.

The resolution was seconded by A. F. Hubbard, Moline, Ill., and was carried by voice vote.

The next subject on the program was a Topical Session on Solar Energy. This was conducted under the chairmanship of Dr. R. C. Jordan, Minneapolis, Minn. The four papers which had been prepared for this session were presented and discussed, following which President Gordon closed the opening session.

SECOND SESSION, TUESDAY, JANUARY 28, 9:30 A.M.

First Vice Pres. E. R. Queer, University Park, Pa., presided at the second session, and H. A. Lockhart, Morton Grove, Ill., as chairman of the Committee on Research, presented the report of that Committee (the complete report can be found on pp. 215–222, of the January, 1958 issue, ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*). Portions of the report are presented here.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1957

Heating, ventilating and air conditioning, now as never before, stand at the threshold of further advance and improvement. New materials are being developed, new methods of accomplishment are being found, and different approaches to old problems are coming to light. Both experimental and analytical research and development are contributing new ideas and methods. Research is the lifeblood of progress, and the SOCIETY through its Research Laboratory in Cleveland, through cooperative research, and through the work of its committees and publications is continuing to serve as a focal point for continuing advancement.

During the fiscal year of the Society, November 1, 1956 to October 31, 1957, an active program of research was carried out at the Laboratory and at the cooperating institutions. Twenty actively-constituted technical advisory committees with a total membership of 266 held 36 meetings during the year. The Committee on Research and its Executive Committee met 8 times.

FINANCES AND EXPENDITURES

Support of the research program comes from 3 sources; the first of these is that portion of the membership dues, amounting to 40 percent which is allocated to research; the second source of funds is the contribution for research from the Exposition; and the third consists of the generous contributions to the program from industrial firms, associations, and individuals. The latter funds are given either for general use or for research along specific lines. Finally, some funds are received for research projects carried out for the United States government or for foundations interested in specific projects of wide general interest. The total expenditures for the year 1956–1957 amounted to $201,623.22.

FACILITIES AND STAFF

The Research Laboratory, located in Cleveland, consists of two buildings, an office building having a floor space of 5300 sq ft, and a research building of 13,000 sq ft. During the year a significant moderization and improvement program was started on the

office building of the Laboratory. The facilities at the Laboratory are very complete and adequate for the type of research which is carried out. The solar calorimetry apparatus, which is particularly well adapted for making solar intensity studies and has recently been used in evaluating the effectiveness of awnings and other shading devices, is housed in a separate building located near one end of the Laboratory's property line.

Work has been intensively under way during the year on a complete rebuilding of the Environment Laboratory. The new facility provides a very flexible test center for carrying out environmental studies under a large number of controlled conditions. Instrumentation for various test projects is continuously being added to and improved with particular emphasis during this year on instrumentation in the field of noise and sound measurement and evaluation.

The large amount of current and reference material received by the library is carefully examined and catalogued for future reference, and the present fine library is an effective adjunct to the research operations.

The research staff consisting of project engineers, assistants, administrative staff, a librarian, secretarial and maintenance staff, ranged from 18 to 22 in number during the year. The number varies with the level of activity under way at any given time, and plans are under way to increase the staff during 1958.

In addition to the research work done at the Laboratory, an extensive program of cooperative research is carried out at a number of university and research laboratories throughout the country. Cooperative research projects are encouraged whenever it is felt that the facilities and available research staff at an outside institution can more effectively and expeditiously carry a research project through to completion than might be done at the Laboratory. The benefits of the cooperative program are extensive as they broaden the base on which research can be built, increase the number of researchers available, and create a greater interest in the problems needing investigation. The work at cooperating institutions is closely followed by the research staff and by members of the Technical Advisory Committees, and the reports of cooperative work are all presented at Society meetings and in Society publications.

Over a period of years the results of research carried out at the Laboratory and at cooperating institutions have become a significant part of the useful literature of the field of heating, ventilating, and air conditioning. Much of this material appears in the annual editions of the HEATING VENTILATING AIR CONDITIONING GUIDE.

In the following parts of this report, detailed summaries of the research activities carried on at the Laboratory and cooperating institutions are presented. These appear as constituent parts of the summaries of activities carried out by each technical advisory committee (TAC).

TECHNICAL ADVISORY COMMITTEE ACTIVITIES

During this year a modification in the procedural practice in relation to membership on the technical advisory committees was presented to the Council by the Committee on Research and approved at the Council meeting held on October 12, 1957. Appointments to technical advisory committees are made by the chairman of the Committee on Research, and in the past the chairman has had to use his best judgment with a minimum of procedural guidance. The new procedures, which are part of Section 5 of the Operational Guide of the Committee on Research, establish a more specific basis on which technical advisory committee membership appointments are made. They make such appointments open to all qualified Society members but provide generally for a 3-year maximum term of service on any specific committee.

AIR CLEANING

Since 1954, the TAC on Air Cleaning has provided advisory guidance for a cooperative study of particle technology being carried on at the University of Minnesota. The purpose of the study is the development of fundamental data on the behavior of airborne

particulate matter. The study has resulted in the development of improved equipment and techniques for obtaining samples of atmospheric dust and determining the size distribution of the dust particles. A reproducible test dust has been developed, along with dust and lint feeding equipment to be used in filter testing.

The TAC on Air Cleaning held 3 meetings during the past year, and reviewed and approved 2 papers resulting from the project at the University of Minnesota.

Air Distribution

The committee held two meetings in 1957 at which plans and programs were discussed and attention was given to the several following research projects in which the Committee is interested.

Case Institute of Technology: A pioneering paper on the performance of radial flow outlets by Professor Koestel was presented at the Semi-Annual Meeting. Professor Koestel has prepared a paper on room air distribution (warm air), and work on a proposed Air Distribution Bulletin should be started soon.

Kansas State College: A final paper on downthrow of heated air jets from various outlets by Helander, Yen and Tripp was presented at the 63rd Annual Meeting. Also a paper on downthrow of heated air from an *ASME* Nozzle by Knaak was presented at the Semi-Annual Meeting in June. Kansas State has submitted to the TAC at its request a research proposal for the determination of heat losses and gains from metal and non-metallic ducts. A Subcommittee headed by Mr. Kennedy is investigating this matter.

University of Buffalo: The TAC is highly interested in research of air flow through duct enlargements undertaken at the University. A progress report on this work has been submitted, and a Subcommittee headed by Mr. Madison is investigating the matter for possible consideration as a cooperative project.

In addition to the subcommittees just mentioned the following are of particular interest:

1. The Subcommittee on Balancing of Air Distribution Systems has prepared a report containing a variety of far-reaching proposals.

2. The Subcommittee on Air Distribution through Perforated Panels has considered a research proposal for further study of this subject.

Important text changes made by the Subcommittee on Guide Review have been incorporated in THE GUIDE 1958, Chapter 31—Air Duct Design.

Combustion

In 1957 the principal interest of the TAC on Combustion was in the research project on Oscillations and Pulsations in Oil and Gas Fired Domestic Heating Equipment, under the guidance of the Pulsation Research Steering Committee.

The solution of the problem of pulsation and resonance in domestic heating equipment, and the determination of methods for eliminating or suppressing such oscillations, is being pursued at Battelle Memorial Institute. Studies indicate that the oscillations are basically of the standing-wave type, and that the principal parameters affecting oscillations are primary air-flow rate for gas furnaces and total air and type of nozzles used for oil furnaces.

A theory has been postulated on the driving mechanism for oscillation in residential gas furnaces equipped with multiple-port, ribbon, or slotted type burners. It is indicated that the general operating conditions leading to oscillations are a function of (*a*) the product of the natural frequency of the burner and the time lag of the burner flame, and (*b*) the ratio of the furnace frequency to the natural frequency of the burner. The extent to which these operating conditions favor oscillation is a function of the ratio of the acoustic driving input from primary combustion to the acoustic damping losses.

An experimental investigation was made to study the conditions under which acoustic interaction could take place between a residential furnace and the room in which it was placed. Methods have been outlined which will help in locating a furnace in a position that will give a low noise level.

Extensive experimental tests are being conducted to determine the required data to complete the theories on the mechanism of oscillation in oil- and gas-fired furnaces, and methods of suppression of such oscillations.

A paper is in preparation on The Fundamentals of Generation of Pulsation in Residential Gas Furnaces With Multiple Port Burners, and four additional technical papers are contemplated.

The project was inaugurated in May 1954 and it is expected that it will terminate in May 1958.

EVAPORATIVE COOLING

The TAC on Evaporative Cooling met 3 times during the year, at the 63rd Annual Meeting in Chicago in February, at the time of the meeting of the Chapters Regional Committee for Region 4 in Los Angeles on May 8, and at the Semi-Annual Meeting 1957 at Murray Bay. During the year, a draft of material on evaporative condensers was prepared for use in THE GUIDE. Similar material was also completed on cooling towers, and new text material on evaporative cooling and humidification is approaching completion.

HEAT PUMP

One meeting of the committee was held in January.

Under the guidance of the Guide Committee, the TAC on the Heat Pump is preparing a new chapter on Heat Pumps for THE GUIDE 1959.

Two papers scheduled for presentation at the 64th Annual Meeting were arranged for by the TAC on the Heat Pump. They are the papers by C. P. Davis, Jr. and R. I. Lipper on Sun-Energy Assistance for Air-Type Heat Pumps, and the one by T. L. Etherington on A Dynamic Heat Storage System.

HEAT TRANSFER THROUGH FENESTRATION

Since 1947 when the Solar Calorimeter at the Laboratory was first put into use, this unique equipment has been in almost constant use, particularly during the summer months. Fifteen important technical papers resulting from this work have already been published, and material for 2 new papers is currently being prepared. Most of the solar heat data currently in use in the field has resulted from these studies.

Shade screens and other slat-type shades have also been evaluated as solar heat barriers. The past summer was devoted to an evaluation of various standard types of canvas and metal awnings, and a first paper on heat gain through windows shaded by canvas awnings has already been prepared by N. Ozisik and L. F. Schutrum of the Laboratory staff.

Future work for the calorimeter is in the planning stage, but roller shades as solar heat barriers are next scheduled for study to be evaluated as solar heat barriers under summer conditions and as radiation barriers under winter conditions. Slat-type shading devices between double glass will also be studied, and it is anticipated that a study of new plastic fenestrations, flat and corrugated reinforced, and in dome or bubble form will also be made.

HEATING AND AIR CONDITIONING LOADS

This committee is organized into subcommittees, with the chairman and vice chairman as ex-officio members of each subcommittee. This arrangement makes it possible to cover more effectively the broad fields of this advisory committee.

The entire Technical Advisory Committee met at the time of the 63rd Annual Meeting. At that time, items of general interest to the committee were discussed. In addition, the committee received a progress report from T. C. Min on field and laboratory studies of infiltration through building entrances. It also received a progress report from the University of Illinois on infiltration studies conducted in two residences.

On July 12, 1957, the subcommittee on Infiltration met at the Laboratory. The manuscript resulting from the research on infiltration through building entrances was reviewed and then was accepted by the subcommittee and recommended for publication. A proposal was also made that further research on infiltration through building entrances be conducted, with emphasis on summer conditions. It was also proposed that an investigation on air infiltration through revolving-door entrances be conducted.

The subcommittee on Heating Loads met on September 27, 1957, in Pittsburgh, Pa. The subcommittee recommended to the Guide Committee that the new weather data developed by H. C. S. Thom on winter design temperatures be incorporated in THE GUIDE 1959, and that certain of these data be included in THE GUIDE 1958. The subcommittee is planning to provide revised material relative to the use of these new data, and also to prepare new information on inside design temperatures.

HOT WATER AND STEAM HEATING

An analytical study on the capacity and sizing of steam piping was completed by W. F. Kerka at the Laboratory. The study showed the values in THE GUIDE to be somewhat overconservative in the smaller pipe sizes with much closer agreement in the larger sizes. Mr. Kerka's work was based on Moody-friction-factor approaches to the problem. However, the unknown effect of condensate in steam pipe lines, which introduces problems of two-phase flow, made it appear desirable to plan an experimental test program. It is believed this will be started early in 1958.

Sizing of condensate receivers has been under study. Very little comprehensive material exists in the literature on this topic.

University of Illinois: Under the direction of W. S. Harris the investigation on the separation of dissolved gases from water in forced circulation hot-water heating systems has been completed. Further work at the University is planned.

University of Florida: The project on the investigation of air lock and air entrainment in hot-water heating systems has been completed.

Northwestern University: The project on noise measurement and noise control in water flow systems has been inactive. It is expected that work will be resumed in early 1958. The metastable research has been completed, and the final report is in preparation.

INDUSTRIAL ENVIRONMENT

The TAC on Industrial Environment held 2 meetings during the year, the first at the 63rd Annual Meeting in February 1957, and the second at the Laboratory in September. The Committee has been working closely with the TAC on Physiological Research in giving guidance for the planning of the environment program at the Laboratory. The Committee has also been active in preparing new data for THE GUIDE and in analyzing the material now contained in THE GUIDE. The Committee strongly feels that the material in THE GUIDE should be strengthened and brought together as a more unified whole, and toward this end chairmen have been appointed to head subcommittees set up to deal with important topics as follows: (1) *Process Aerodynamics*—W. C. L. Hemeon, Chairman; (2) *Building Structures*—H. E. Ziel, Chairman; (3) *Air Contaminants*—A. D. Brandt, Chairman; (4) *Controlled Air Supply*—J. H. Clarke, Chairman; and (5) *Industrial Exhaust Systems and Air Cleaning*—J. M. Kane and R. P. Warren, Co-Chairmen.

The Committee, in its realization that not enough material on industrial ventilation is appearing in the literature, is considering the idea of arranging a number of articles for possible use in the technical press.

INSULATION

The TAC on Insulation has completed work on recommendations for two chapters in THE GUIDE, Chapter 9 on Heat Transmission Coefficients of Building Materials (complete revision); and Chapter 27 on Pipe and Industrial Insulation (complete revision). Follow-up work in connection with these chapters as well as Chapter 10 on Moisture in Building Construction has been done.

New work, which is being undertaken, involves:

1. Preparation of suggested material for a new chapter on *Good Practice Recommendations for Insulation.*

2. Establishment of task forces in specific fields which will work towards inauguration of research on other projects such as:

(*a*) Study of attic and crawl space ventilation, air and surface temperatures, and related conditions, so that their effect on heat transmission coefficient of building sections and on the control of condensation can be evaluated. This study should include both summer and winter conditions.

(*b*) Recommendations for heat and moisture control for slab-on-grade construction.

(*c*) Moisture in insulated built-up roofs; effect of moisture on thermal conductivity.

(*d*) Loose-fill insulations, including sprayed-on and foamed-in-place insulations.

ODORS

Apparatus and a method of precisely odorizing fabric samples has been developed. A *loading* tank can be kept at a constant level of any selected liquid odorant and at a constant temperature and humidity. The rate of odorant release from fabric samples has been determined by a dilution method in which rows of clean bottles are odorized by successively smaller fabric samples until one is found which results in threshold odor in a bottle. Rows of bottles are loaded with samples which have been aired under controlled conditions for successively longer times.

It has been found that fabric samples lose odor rapidly for a few minutes, moderately fast for a few hours, and then slowly for many hours. The three distinct time constants can be explained by analogy. If water loss were being measured, starting with samples taken from liquid:—weight loss would be rapid as water held in the weave dripped out; weight loss would next be less rapid as fiber surfaces dried; and then would settle to a slow rate as moisture worked it way out of fiber interiors.

An alternative method of measuring fabric odor level is now being evaluated.

A subcommittee is studying the possible application of gas chromatography to this problem. This is an attempt to anticipate difficulties with odor blends. Samples can be expected to release individual odors at different rates, rendering the bottle or syringe method nearly useless.

PHYSIOLOGICAL RESEARCH AND SENSATIONS OF COMFORT

Activities of these 2 technical advisory committees are so closely related in connection with the planning of the Environment Test Room that a common report for the 2 committees is presented.

In 1956 activity and planning for a complete rebuilding of the Environment Test Room were started, and this rebuilding is approaching completion. It is expected that the Test Room will go into use with human subjects in early 1958. Temperature, humidity, air motion, along with radiant wall temperatures, can be varied in the room over a wide range of conditions. The environment center is also provided with 2 pre-test rooms in which the subjects can be acclimatized to specified equilibrium conditions before they enter the Environment Room. An observation room is also provided for the operating staff.

The project will be conducted as a series of programs essentially as prepared with the cooperation of the technical advisory committees by Nathaniel Glickman, consulting physiologist on the project.

Program I. Influence of the Thermal Environment on the Comfort of Sedentary Workers. This program will be carried out first, primarily to re-evaluate certain of the data and values and their relative allocation on the comfort chart of the Society.

Program II will extend the objectives of Program I and introduce variations of radiant wall temperatures, so that the effect of symmetrical as well as asymmetrical radiation can be studied over a wide variety of conditions.

Program III. Optimum Conditions for Comfort at Different Levels of Work. This program will endeavor to determine the desirable comfort ranges at varying work levels.

Program IV. Human Productivity or Occupational Efficiency. This is one of the more difficult researches to carry out, as it will have to measure the effects of environment on productivity. It is thought to be axiomatic that an occupant in an uncomfortable environment not well adapted to his physiological needs will perform less effectively than a person in a more suitable environment.

Program V. The Comfort of School Children as Related to Learning Rates.

These several programs will necessarily take many years to complete. The end results can be extremely far reaching and of great significance.

Plant and Animal Husbandry

The TAC met twice during the year. A number of research projects that might profitably be pursued were considered, but it was realized that the literature is so widely scattered that a careful search of all related literature should be made as a first step in broadly planning a program of action. During the year the ASHAE Council approved the formation of a Joint Committee with the *American Society of Agricultural Engineers*. It is believed that the new Joint Committee might be very effective, first in correlating the available knowledge of this field, and second in helping plan directions in which research should be carried out.

Solar Energy Utilization

The TAC on Solar Energy Utilization met during the 63rd Annual Meeting in February 1957. Among items considered was a proposal that a separate chapter on solar energy be prepared for THE GUIDE. Another important activity of the committee during the past year has been the preparation of a summary article on Solar Energy Utilization for Heating, Cooling, Distillation, and Drying. This article was completed during the year.

University of Minnesota: Work is continuing at the University of Minnesota on the cooperative project on solar energy collectors. Data on the amount of heat absorbed, type of radiation and transmittance of single and double glass are being measured. This work will continue over a period of time. However, significant data have already been obtained.

Sorption

The TAC on Sorption held 2 meetings during the year, one at the 63rd Annual Meeting, February 1957 and the second at the Semi-Annual Meeting in June. The committee is actively following progress of its 2 research projects, and in addition is in process of developing a tentative standard, a *Proposed Method for Testing Sorption-Type Dehumidifiers*.

The Pennsylvania State University: The cooperative research project is being conducted on sterilization of air by solid sorbents.

University of Toledo: A cooperative research project is in progress on the sterilization of air by liquid sorbents.

As both of these cooperative projects have been under way for but a short time, comprehensive progress reports are not available.

SOUND AND VIBRATION CONTROL

The Committee is directing its activities through 3 subcommittees: (1) Subcommittee on Measurement of Sound-Power Level of a Fan, R. D. Madison, Chairman; (2) Subcommittee on Attenuation of Sound in Duct Systems, C. M. Ashley, Chairman; (3) Subcommittee on Review of Guide Chapter on Sound Control, J. B. Chaddock, Chairman.

Through frequent meetings the committee has been guiding the sound-study program at the Laboratory.

At the Semi-Annual Meeting, Murray Bay, June 1957, the Committee gave its active support to the organization of a Symposium on Sound and Vibration. Ten papers on the subject were presented during the Symposium and at technical sessions.

A meeting of the subcommittee on Attenuation of Sound in Duct Systems was held to discuss the sound program now being conducted by the Research Laboratory and sponsored by the U. S. Navy, Bureau of Ships.

A meeting of the subcommittee on the Measurement of Sound-Power Level of a Fan was also held at Murray Bay to discuss proposed joint activities of the ASHAE and *ASRE* toward developing a standard for measuring the sound output of air-conditioning equipment. F. O. Urban, member of the *ASRE*, was present at the subcommittee meeting.

In February 1957 the Council authorized the Standards Committee of the Society to appoint representatives to a joint committee of ASHAE and *ASRE* on equipment noise testing. The three ASHAE members of the joint committee, Messrs. Ashley, Hubbard and Madison are all presently members of the TAC, while R. J. Wells, also a member of the TAC, is consultant to the joint committee.

THERMAL CIRCUITS

The thermal-circuit technique is a powerful analytical method of synthesizing the thermo-physical characteristics of building components into the performance of the structure as a whole. Pioneered by H. B. Nottage in 1954, this method was originally guided by the TAC on Cooling Loads. In 1956, the TAC on Thermal Circuits was formed to develop and validate methods of determining the behavior of complex thermal systems and to formulate application engineering data in cooperation with other TACs.

Ten ASHAE research papers have been presented to date.

As a result of the continuing cooperative project with University of California at Los Angeles, H. A. Buchberg has prepared 2 papers.

The TAC has formulated, and recommended to the Committee on Research, a program of additional analytical and experimental work having the objective of eventually providing readily-usable design and application engineering data.

WEATHER DATA

The TAC on Weather Data extended its studies in connection with summer dry-bulb and summer wet-bulb information. A similar study was completed by the Weather Data Committee on winter dry-bulb temperatures and presented to the Society in the form of a paper by H. C. S. Thom.

The TAC also reviewed a paper prepared by W. L. Holladay, a member of the committee. This paper, entitled *Local Climatic Weather Data* is to be presented at the 64th Annual Meeting, Pittsburgh, January, 1958.

Also during this past year, the TAC on Weather Data has made a preliminary investi-

gation into the dry-bulb and wet-bulb data for summer. This preliminary investigation has shown some startling deviations from the existing data in THE GUIDE. Members of the Committee have been most diligent in furnishing their suggestions, criticisms, and studies of the complex problems associated with these studies.

The next order of business was the report of the Inspectors of Election, covering the balloting for officers, members of the Council, members of the Committee on Research, and on amendments to Article II and Article VII of the By-Laws. Mr. Hach presented the report as follows.

REPORT OF INSPECTORS OF ELECTION

Ballot for Officers, 1958:	Total
President, E. R. Queer, University Park, Pa.	2530
First Vice President, A. J. Hess, Los Angeles, Calif.	2529
Second Vice President, Walter A. Grant, Syracuse, N. Y.	2526
Treasurer, J. H. Fox, Toronto, Ont., Canada	2520
Members of Council (three-year term)	
F. H. Faust, Montclair, N. J.	2497
Fred Janssen, Denver, Colo.	2502
James W. May, Louisville, Ky.	2491
G. B. Priester, Baltimore, Md.	2494
Committee on Research (three-year term)	
P. R. Achenbach, Washington, D. C.	2495
S. F. Gilman, Syracuse, N. Y.	2490
N. B. Hutcheon, Ottawa, Ont., Canada	2492
R. M. Stern, Seattle, Wash.	2492
H. E. Ziel, Detroit, Mich.	2482
TOTAL BALLOTS RECEIVED	2607
TOTAL LEGAL BALLOTS	2542
INVALID BALLOTS	65

Scattering votes: Officers, 5; Members, 7; and Committee on Research, 10.

Amendments to By-Laws	Total Yes	Total No
1. Art. II *Sec.* 3 Qualifications	2531	11
2. Art. II *Sec.* 3 (a) Honorary Member	2529	13
3. Art. II *Sec.* 3 (b) Presidential Member	2533	9
4. Art. II *Sec* 3 (c) Life Member	2535	7
5. Art. II *Sec.* 3 (d) Fellow	2519	23
6. Art. II *Sec.* 3 (e) Member	2531	11
7. Art. II *Sec.* 3 (f) Associate Member	2534	8
8. Art. II *Sec.* 3 (g) Affiliate	2530	12
9. Art. II *Sec.* 3 (h) Student	2534	8
10 Art II *Sec* 4 Elections	2531	11
11. Art. II *Sec.* 5 Proposals and Applications	2532	10
12. Art. II *Sec* 6 Rights and Privileges	2537	5
13. Art. VII *Sec.* 2 Council Committees	2535	7

Respectfully submitted,

E. C. HACH, *Chairman*
J. H. ALLISON
E. C. SMYERS

Note: Since these By-Law amendments had thus been approved by the balloting, they are incorporated in the By-Laws as printed in full elsewhere in this Volume.

Four papers had been programmed for this session, one being by title only. Following their presentation and discussion, 1st Vice Pres. Queer adjourned the session.

THIRD SESSION, TUESDAY, JANUARY 28, 9:30 A.M.

A panel of 5 speakers assembled for this Symposium on High-Temperature Water. Second Vice Pres. A. J. Hess, Los Angeles, Calif., presided and P. N. Vinther, Dallas, Tex., was moderator. The 5 papers can be found substantially in full in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning.*

Economic Evaluation of High-Temperature Water, by E. M. Thompson is in the April, 1958 issue on p. 140.

Marine Application of High-Temperature Water, by S. W. Brown is in the March issue on p. 161.

Design of High-Temperature Water Systems for Military Installations, by C. A. Carter and B. L. Sturdevant is in the February, 1958 issue on p. 109.

Heating & Air Conditioning a Civilian Airport, by Charles Broder is in the March 1958 issue on p. 147.

British and European Design and Construction Methods, by George Applegate, Jr., is in the March 1958 issue on p. 169.

These 5 symposium papers are not printed in the TRANSACTIONS, but are available from the Society in full together with the discussions, as a separately printed Bulletin.

FOURTH SESSION, WEDNESDAY, JANUARY 29, 9:30 A.M.

Pres. P. B. Gordon presided at the fourth session during which 3 programmed papers were presented and discussed.

FIFTH SESSION, WEDNESDAY, JANUARY 29, 9:30 A.M.

This session was called to order by Treas. C. H. Pesterfield, E. Lansing, Mich. There were 3 papers, which were presented and discussed.

SIXTH SESSION, WEDNESDAY, JANUARY 29, 2:00 P.M.

At this session, which was presided over by 1st Vice Pres. E. R. Queer, 3 technical papers were presented and discussed.

SEVENTH SESSION, WEDNESDAY, JANUARY 29, 2:00 P.M.

This session was a Symposium on School Heating, Ventilating and Air Conditioning, with Pres. P. B. Gordon presiding. W. G. Hole, Montreal, Que., Canada, acted as moderator. Five papers were prepared for this symposium and each was presented by its author.

One of these papers, entitled HV&AC Design Practice for Schools, by Henry Wright can be found in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, February, 1958 issue p. 137.

Three others, namely: Educator's View of Need for Good Classroom Environment, by G. B. Wadzeck; Economics of School Heating and Air Conditioning, by Z. A. Marsh; and State Board Problems—Financing and Others, by C. B. Hershey, appear in the April, 1958 issue of the JOURNAL SECTION.

The fifth paper Looking into the Future by E. C. Good, appears in shortened form in the May, 1958 issue of the JOURNAL SECTION.

These symposium papers are not printed in TRANSACTIONS but are available from the Society in the form of a separately printed Bulletin, containing the papers and the discussions.

Following the completion of the symposium session, President Gordon called for the report of the Committee on Resolutions. It developed that two such committees had been appointed. The deliberations of the two committees brought forth the following joint

RESOLUTIONS

WHEREAS, for sixty-three annual meetings and sixty-two semi-annual meetings your Resolutions Committees have used so many *whereases*, your present *whereas Committee* is herewith abandoning the use of the word *whereas*;

BECAUSE, the 64th Annual Meeting of the ASHAE, held in the city of Pittsburgh, whose dreams and schemes come true, is drawing to a close,

BECAUSE, the Pittsburgh Chapter and its Committee on Arrangements have provided so well for our comfort and pleasure from the Gay Nineties to the atomic age;

BECAUSE, Ted Rockwell, general chairman, isn't as worried as he appears to be,

BECAUSE, the arts and sciences of our profession have been advanced by the excellent technical sessions and symposia so well conducted under the direction of the Officers, staff and committees, and

BECAUSE we are also tired of the phrase *Be It Resolved*, let it here now

BE DECIDED, that our congratulations and thanks are due to

The members of our Society honored by the grade of FELLOW—namely, A. L. Baum, the late A. C. Buensod, I. W. Cotton, E. L. Crosby, Prof. F. B. Rowley, E. P. Heckel, Prof. C. F. Kayan and G. M. Simonson,

George L. Tuve, recipient of the F. Paul Anderson Medal,

Linn Helander, honored by the first presentation of the Life Members Club,

Wayne E. Springer, who received the ASHAE-Homer Addams Award,

Ben Gordon and his charming wife, Alice, for being a wonderfully competent presidential team during the past year,

Mayor David L. Lawrence for actually coming to the Welcome Luncheon instead of sending an aide, and for telling us how through civic cooperation such dreams as the Golden Triangle can come true,

Allison R. Maxwell, Jr., whose talk at our Annual Banquet we look forward to with eager anticipation,

The Pittsburgh Chapter, its officers and committees,

R. B. Stanger, honorary chairman, T. F. Rockwell, Jr., general chairman, D. W. Loucks, vice chairman,

Mrs. B. B. Reilly and the ladies of her committee,

The hotel management and staff for taking good care of us (except electronically),

The headquarters staff for its continued efficient handling of all the myriad details involved in our meetings,

To Ben Gordon for appointing not one, but two Resolutions Committees.

Respectfully submitted,

J. W. MAY—*Chairman* of *one*
RESOLUTIONS COMMITTEE
C. M. BURNAM, JR. HARRY G. GRAGG
E. L. CROSBY D. LORNE LINDSAY

Respectfully submitted,

E. L. CROSBY, *Chairman* of the
other RESOLUTIONS COMMITTEE
C. M. BURNAM, JR. D. LORNE LINDSAY
Harry G. GRAGG J. W. MAY

These resolutions were antiphonally presented by the two committee chairmen and, on motion made and seconded, President Gordon declared their adoption.

Following adoption of the resolutions, and there being no unfinished or new business, President Gordon adjourned the meeting with the statement that the installation of the newly elected officers and the new members of the Council would take place during the Annual Banquet.

PROGRAM—64th ANNUAL MEETING

Penn-Sheraton Hotel, Pittsburgh, Penna.

January 27–29, 1958

Saturday—January 25

9:00 a.m. Executive Committee (*Parlor C*), E. R. Queer, *Chairman*
9:30 a.m. Program and Papers Committee (*Parlor E*), John Everetts, Jr., *Chairman*
1:30 p.m. Regions Central Committee (*Parlor B*), A. J. Hess, *Chairman*
2:00 p.m. Research Executive Committee (*Parlor D*), H. A. Lockhart, *Chairman*
4:30 p.m. Finance Committee (*Parlor C*), Walter A. Grant, *Chairman*

Sunday—January 26

10:00 a.m. Council Meeting (*Parlors B & C*)
10:00 a.m. Committee on Research (*Parlors E & F*), H. A. Lockhart, *Chairman*
1:30 p.m. REGISTRATION (*Fort Duquesne Room*)

Monday—January 27

9:00 a.m. REGISTRATION (*Fort Duquesne*)
9:00 a.m. FIRST SESSION (*Urban*)
Call to Order—Pres. P. B. Gordon
Report of President—P. B. Gordon
Report of Treasurer—C. H. Pesterfield
Amendments to By-Laws

TOPICAL SESSION ON SOLAR ENERGY

R. C. Jordan, *Chairman*

Direct Solar Radiation Available on Clear Days, by J. L. Threlkeld and R. C. Jordan, Minneapolis, Minn., presented by Mr. Threlkeld.

A High-Flux Low-Temperature Solar Collector, by R. G. Nevins, Manhattan, Kans., and P. E. McNall, Jr., Minneapolis, Minn., presented by Mr. Nevins.

Performance of a Solar Heated Office Building, by F. H. Bridgers, D. D. Paxton and R. W. Haines, Albequerque, N. M., presented by Mr. Bridgers.

Sun Energy Assistance for Air-Type Heat Pumps, by C. P. Davis, Jr. and R. I. Lipper, Manhattan, Kans., presented by Mr. Davis.

12:30 p.m. WELCOME LUNCHEON

Toastmaster: T. F. Rockwell, Pittsburgh

Invocation: Rev. Vernon F. Gallagher, C.S. Sp., president, Duquesne University

Speaker: Hon. David L. Lawrence, Mayor of Pittsburgh

Subject: Pittsburgh, A City Whose Dreams Come True

Presentation of Certificates to Fellows: A. L. Baum, New York, N. Y.; A. C. Buensod, New York, N. Y.; C. F. Kayan, New York, N. Y.; I. W. Cotton, Indianapolis, Ind.; E. L. Crosby, Baltimore, Md.; E. P. Heckel, Chicago, Ill.; F. B. Rowley, Minneapolis, Minn.; and G. M. Simonson, San Francisco, Calif.

Presentation of ASHAE-Homer Addams Award

2:00 p.m. Inspection Trips for Ladies and Men to H. J. Heinz Co., Buhl Planetarium, Tour of Pittsburgh

2:00 p.m. TAC on Air Cleaning (*Parlor B*), E. F. Snyder, Jr., *Chairman*

2:00 p.m. TAC on Heat Transfer through Fenestration (*Parlor C*), R. W. McKinley, *Chairman*

2:00 p.m. TAC on Hot Water and Steam Heating (*Parlor D*), M. W. McRae, *Chairman*

2:00 p.m. TAC on Sorption (*Parlor E*) G. L. Simpson, *Chairman*

2:30 p.m. Ladies' Tea and Entertainment (*Monongahela*)

9:30 p.m. Gay Nineties Party

Tuesday—January 28

9:00 a.m. REGISTRATION (*Fort Duquesne*)

9:30 a.m. SECOND SESSION (*Urban*)

Call to Order—1st Vice Pres. E. R. Queer

Report of Committee on Research—H. A. Lockhart, *Chairman*

Report of Inspectors of Election

Cooling Load from Thermal Network Solutions, by Harry Buchberg, Los Angeles, Calif., presented by S. F. Gilman, Syracuse, N. Y.

The ASHAE Air-Borne Dust Survey, by K. T. Whitby, A. B. Algren, R. C. Jordan and J. C. Annis, Minneapolis, Minn., presented by Mr. Whitby.

Natural Convection Cooling and Dehumidifying, by L. G. Seigel, Erie, Penna. and W. L. Bryan, Cleveland, Ohio, presented by Mr. Seigel.

Stable Hot-Wire Anemometer for Low Speeds, by J. F. Kemp, Pretoria, South Africa (by title only).

9:30 a.m. THIRD SESSION (*Ballroom*)

Call to Order—2nd Vice Pres. A. J. Hess

SYMPOSIUM ON HIGH-TEMPERATURE WATER

P. N. Vinther, *Moderator*

Economic Evaluation of High Temperature Water, by E. M. Thompson

Marine Application of High Temperature Water, by S. W. Brown

Design of High Temperature Water Systems for Military Installations, by C. A. Carter and B. L. Sturdevant.

Heating and Air Conditioning a Civilian Airport, by Charles Broder

British and European Design and Construction Methods, by George Applegate, Jr.

Report of Inspectors of Election

10:00 a.m. Ladies Bus Tour and Luncheon—Ft. Pitt Block House, Pittsburgh's Mt. Washington, University of Pittsburgh; dining at University Club

12:00 noon Life Members' Dutch Treat Luncheon (*Parkview*)

1:00 p.m. Inspection Trips for Men Only to Homestead Steel Works, Pittsburgh Consolidated Coal Co., Gulf Oil Corp. Research Laboratories

1:30 p.m. Inspection Trips for Ladies and Men to Buhl Planetarium, Tour of Pittsburgh

1:30 p.m. Nominating Committee (*Club Room*)

2:00 p.m. TAC on Heat Pump (*Parlor B*), F. R. Ellenberger, *Chairman*

2:00 p.m. TAC on Combustion (*Parlor C*), H. R. Limbacher, *Chairman*

2:00 p.m. TAC on Industrial Environment (*Parlor D*), K. E. Robinson, *Chairman*

2:00 p.m. TAC on Solar Energy Utilization (*Parlor E*), R. C. Jordan, *Chairman*

2:00 p.m. TAC on Weather Data (*Parlor F*), John Everetts, Jr., *Chairman*

2:00 p.m. TAC on Physiological Research (*Sky Room*), M. K. Fahnestock, *Chairman*

7:00 p.m. Past Presidents' Dinner (*Room 1690*)

7:30 p.m. TAC on Evaporative Cooling (*Parlor F*), Leo Hungerford, *Chairman*

7:30 p.m. TAC on Heating and Air Conditioning Loads (*Parlor E*), H. T. Gilkey, *Chairman*

7:30 p.m. TAC on Insulation (*Parlor B*) M. W. Keyes, *Chairman*

7:30 p.m. TAC on Odors (*Parlor C*) A. B. Hubbard, *Chairman*

7:30 p.m. TAC on Sound and Vibration Control (*Parlor D*), A. F. Hubbard, *Chairman*

8:00 p.m. Chapter Editors' Conference (*Parkview*)

Wednesday—January 29

9:00 a.m. REGISTRATION (*Fort Duquesne*)

9:30 a.m. FOURTH SESSION (*Ballroom*)

Call to Order—Pres. P. B. Gordon

Effect of Heated-Floor Temperatures on Comfort, by R. G. Nevins and A. O. Flinner, Manhattan, Kans., presented by Mr. Nevins.

Lighting and Cooled Air Effects on Panel Cooling, by L. F. Schutrum and T. C. Min, Cleveland, Ohio, presented by Mr. Min.

Radiant Drafts from Cold Ceilings, by H. E. Ronge and B. E. Lofstedt, Uppsala, Sweden, presented by C. M. Humphreys, Cleveland, Ohio.

9:30 a.m. FIFTH SESSION (*Urban*)

Call to Order—Treas. C. H. Pesterfield

Local Climatic Weather Data, by W. L. Holladay, Los Angeles, Calif., presented by A. J. Hess, Los Angeles, Calif.

A Dynamic Heat Storage System, by T. L. Etherington, Schenectady, N. Y., presented by Mr. Etherington.

Improving Attic Space Insulating Values, by F. A. Joy, University Park, Penna., presented by Mr. Joy.

1:00 p.m. Luncheon and Entertainment (*Monongahela*)

2:00 p.m. SIXTH SESSION (*Urban*)

Call to Order—1st Vice Pres. E. R. Queer

Chimney and Stack Design for Gas-Fired Equipment, by R. L. Stone, Belmont, Calif., presented by Mr. Stone.

Pressure Losses and Flow Characteristics of Multiple-Leaf Dampers, by E. J. Brown and J. R. Fellows, Champaign, Ill., presented by Mr. Brown.

Fan Noise Variation with Changing Fan Operation, by R. D. Madison and J. B. Graham, Buffalo, N. Y., presented by Mr. Madison.

2:00 p.m. SEVENTH SESSION (*Ballroom*)

Call to Order—Pres. P. B. Gordon

SYMPOSIUM ON SCHOOL HEATING, VENTILATING AND AIR CONDITIONING

W. G. Hole, *Moderator*

Educator's View of Need for Good Classroom Environment, by G. B. Wadzeck.

HV&AC Design Practice for Schools, by Henry Wright.

Economics of School Heating and Air Conditioning, by Z. A. Marsh.

State Board Problems—Financing and Others, by C. B. Hershey.

Looking Into the Future, by E. G. Good.

Report of Resolutions Committee

Unfinished Business

New Business

Adjournment

6:00 p.m. Social Hour (*Urban*)

7:00 p.m. ANNUAL BANQUET (*Ballroom and Urban*)

Toastmaster: Thomas Pratt: Member of Ingham, Boyd & Pratt, Architects

Invocation: Dean N.R.H. Moor, D.D., Trinity Episcopal Cathedral, Pittsburgh

Speaker: Allison R. Maxwell, Jr. President, Pittsburgh Steel Co.

Subject: What Steel Stability Means To You

Introduction and Installation of 1958 Officers

Presentation of F. Paul Anderson Medal to G. L. Tuve by Pres. P. B. Gordon

Presentation of Past President's Award to P. B. Gordon by John W. James

10:00 p.m. Dancing

Thursday—January 30

10:00 a.m. Organization Meeting of 1958 Council (*Parlors B & C*)

10:00 a.m. Organization Meeting of 1958 Committee on Research (*Parlors E & F*)

COMMITTEE ON ARRANGEMENTS

T. F. Rockwell, *General Chairman*

D. W. Loucks, *Vice Chairman*

R. B. Stanger, *Honorary Chairman*

Finance—H. Lee Moore, *Chairman*; P. A. Edwards, W. D. Simpson, E. C. Smyers, K. W. Stickle.

Hospitality—E. J. Busch, *Chairman*; W. A. Allen, Sanford Bausman, R. J. Fitzgerald, W. E. Keist, R. M. Meucci, L. S. Pelley, H. H. Reich, A. E. Summer.

Inspection Trips—A. F. Metzger, *Chairman*; H. A. Biber, J. L. Coleman, Jr., E. C. Hach, A. F. Nass, Jr., J. F. Sasser. C. H. Schneider.

Ladies Activities—Mr. and Mrs. B. B. Reilly, *Co-Chairmen*; Mmes. W. A. Allen, J. H. Allison, J. C. Benson, E. J. Busch, G. M. Comstock, D. E. Hickey, D. B. Hicks, J. Hough, J. J. Kelly, D. W. Loucks, H. W. Lutz, G. J. Parros, H. H. Reich, J. N. Riley, E. H. Riesmeyer, Jr., T. F. Rockwell, C. H. Schneider, A. C. Schock, G. E. Smetak, R. M. Toucey.

Public Relations—B. R. Small, *Chairman*; J. N. Riley, *Vice Chairman*, J. C. Benson, M. J. Hannah, I. C. Laux, E. A. Smith, R. J. J. Tennant.

Reservations—D. B. Hicks, *Chairman*; J. G. Beres, Edward Claitman, J. F. Fleming, R. C. Firsching, J. H. Llewellyn, Charles Mitchell, R. A. Roos, L. G. Russell, Harold Schratter, C. W. Stanger, R. M. Toucey, L. B. Wilkes.

Sessions and Facilities—C. W. Stanger, *Chairman*; A. W. Marshall, *Vice Chairman, charge of Facilities*; C. L. Benn, Walter Lambert, Jr., E. C. Smyers.

1621

No. 1621

CHARTER

●

Provisions of

CERTIFICATE OF INCORPORATION

of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

(As filed on January 24, 1895 and amended by Certificates of Amendment filed on or about May 20, 1914, March 8, 1946, December 7, 1949, February 1, 1952 and December 8, 1954.)

1. The corporate name of the Society is to be known as AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

2. The certificate of incorporation was filed in the office of the Secretary of State on or about January 24, 1895. Amended certificates of incorporation were filed in the office of the Secretary of State on or about May 20, 1914, March 8, 1946, December 7, 1949, February 1, 1952 and December 8, 1954.

3. The number of directors shall be not less than thirteen (13) nor more than twenty-one (21). The Board of Directors shall be styled "The Council."

4. The purposes for which this corporation is organized are:

(*a*) To advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, the results of which shall be made freely available to the public.

(*c*) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, reports, educational programs and scientific and technical data relating to heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*d*) To engage in educational activities (not including the conduct of any school, schools or institutions of learning), and to encourage the adoption and maintenance of high standards of instruction and technical and professional training in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*e*) To cooperate with governmental agencies and with universities, colleges, schools and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants and awards in furtherance of the foregoing purposes.

(*f*) To assist in the formation of local chapters and student and other branches, and to regulate, operate and control the same under the direction and at the pleasure of the corporation, but no local chapter or branch shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(*g*) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(*h*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*i*) This corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of this corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the corporation, the Council (directors) shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer or director of the corporation.

5. The principal place of business or office of said Society is to be in the City and County of New York.

(*a*) The territory in which the corporation's operations are principally to be conducted is in all parts of the United States and its possessions and in the Dominion of Canada, but the corporation may on occasions extend its operations to other parts of the world.

(*b*) No officer, director or member of this corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as the Board of Directors may determine.

(*c*) Each director of the corporation shall be indemnified by the corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit or proceeding to be liable for willful negligence, misfeasance or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote or otherwise.

CONTENTS OF BY-LAWS

BY-LAWS
of
American Society of Heating and Air-Conditioning Engineers, Inc.

• • •

ARTICLE I

Organization

Section 1. ***Organization.*** American Society of Heating and Air-Conditioning Engineers, Inc. is a membership corporation, organized and existing under and by virtue of the laws of the State of New York.

Section 2. ***Object.*** The object of the Society is to advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

Section 3. ***Members.*** The rights and privileges of a member of the Society shall be personal to himself and shall not be delegated or transferred, except that each member entitled to vote may vote in person or by proxy as hereinafter provided.

Section 4. ***Government.*** The Society shall be governed by its Charter and these By-Laws, the Rules promulgated by the Council in harmony therewith, and any amendments to the foregoing.

ARTICLE II

Memberships

Section 1. ***Grades.*** The grades of membership in the Society shall be designated as follows: (*a*) Honorary Members, (*b*) Presidential Members, (*c*) Life Members, (*d*) Fellows, (*e*) Members, (*f*) Associate Members, (*g*) Affiliates, and (*h*) Students.

Section 2. ***Construction.*** Whenever in these By-Laws the words "MEMBER" or "MEMBERS" appear in upper-case letters (*sic*, "MEMBER," "MEMBERS"), they shall be taken to be synonymous with the terms "Honorary Members," "Presidential Members," "Life Members," "Fellows," and "Members"; and whenever the words "member" or "members" appear in lower-case letters (*sic*, "member," "members") and the context permits it, they shall be taken to be synonymous with all the grades of membership in the Society.

Section 3. ***Qualifications.***

(*a*) *Honorary Member.* An Honorary Member shall be a notable person of preeminent professional distinction.

(*b*) *Presidential Member.* A Presidential Member shall be a Past President of the Society.

(*c*) *Life Member.* A Life Member shall be a Member, an Associate Member, or an Affiliate, in good standing, who has attained the age of sixty-five (65) years and has paid dues in said grades for thirty (30) years.

(*d*) *Fellow.* A Fellow shall be a MEMBER who has attained unusual distinction in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, or in the teaching of major courses in said arts and sciences, or who by reason of invention, research, original work, or as an engineering executive on projects of unusual or important scope, has made substantial contribution to said arts and sciences, and who has attained the age of forty-five (45) years, and has been in good standing as a MEMBER for a period of at least ten (10) years prior to the date of his proposal for Fellow grade.

(*e*) *Member.* A Member shall be an engineer or teacher in engineering having ten (10) years or more of experience of a character satisfactory to the Council in the arts relating to the sciences of heating, ventilating. cooling or air conditioning, or the allied arts and sciences, of which four (4) years or more shall have been in responsible charge of important engineering work or in the teaching of courses in said arts and sciences; or an architect, chemist, physician, scientist or other person, deemed by the Council to be qualified by reason of special experience in the said arts and sciences.

Graduation from an engineering school with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from an engineering school outside of the United States maintaining similar standards, or from an engineering school of collegiate grade approved by the Council shall be deemed equivalent to four (4) years of such experience; from other technical schools maintaining four (4) year courses, three (3) years; from a non-engineering college, two (2) years; and each successfully completed year in an engineering college shall be deemed equivalent to one (1) year of such experience.

(*f*) *Associate Member.* An Associate Member shall be a person having been graduated from a college or school of engineering with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from an engineering school outside of the United States maintaining similar standards, or from an engineering school of collegiate grade approved by the Council, or possessing eight (8) years of experience satisfactory to the Council in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences. Each successfully completed year in such college or school shall be deemed equivalent to one (1) year of such experience.

(*g*) *Affiliate.* An Affiliate shall be a person not possessing qualifications for Member or Associate Member grades but whose pursuits, scientific attainments or practical experience qualify him to cooperate with heating, ventilating, cooling or air conditioning engineers in the advancement of engineering knowledge and practice.

(*h*) *Student.* A Student shall be a person registered as a graduate student or as an under-graduate student pursuing an engineering curriculum accredited by the Engineers' Council for Professional Development or in a college or school of collegiate grade approved by the Council. A student membership shall terminate upon the student's graduation, or at the close of the school term during which his enrollment as a student has ceased.

Section 4. ***Elections.*** Membership in the Society and advancement in membership grade shall be by vote of the Council on proposals or applications, or as set forth below. An unanimous vote by secret ballot shall be required for election to Honorary Member grade. Fifteen (15) votes by secret ballot shall be required for election to Fellow. A president of the Society shall advance to Presidential Member upon installation of his successor. A two-thirds vote of the Council shall be required for election to other membership grades.

A person whose student membership has terminated may be advanced to Associate Member or Affiliate grade by the Council upon application providing proof of his qualifications therefor.

The grade of Honorary Member shall be conferred on no more than three (3) persons in any calendar year. The grade of Fellow shall be conferred on no more than ten (10) MEMBERS in any calendar year. No member of the Council shall be elected to Honorary Member or Fellow grade.

Before election or advancement to Affiliate, Associate Member, or Member, the name of the applicant shall be published in an issue of the JOURNAL or mailed to all members.

Section 5. ***Proposals and Applications.*** Proposals and applications shall be on forms approved by the Council. Proposals for Honorary Member and Fellow grades shall be signed by no less than ten (10) MEMBERS, no more than five (5) of whom shall be members of the same Chapter of the Society. Members of the Council may not be proposers. No application shall be required for advancement to Life Member.

Section 6. ***Rights and Privileges.*** MEMBERS and Associate Members shall have the right to vote in the Society. MEMBERS shall be eligible to hold elective office in the Society.

Section 7. ***Resignation.*** Any member may resign his membership by writing to the Executive Secretary, who shall present the resignation to the Council at its next meeting. If the dues of the resigning member shall have been paid to the end of the quarter-annual period immediately preceding the date of the receipt of resignation, his resignation shall be accepted.

Section 8. ***Suspension and Expulsion.***

(*a*) *Non-Payment of Dues:* If any Fellow, Member, Associate Member, or Affiliate shall fail to pay his dues by April 1, he shall be classed as delinquent and not entitled to vote; if such dues are not paid by July 1, he shall be classed as not in good standing and his membership shall be suspended; if such dues are not paid by December 1, the Executive Secretary shall notify the suspended member by registered mail that unless such dues are paid by December 31, he shall cease to be a member of the Society, and upon his failure to cure such default by December 31, his membership in the Society shall cease; if any Student shall fail to pay his dues by December 31, the delinquent Student's membership shall cease and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased; provided that upon written application satisfactorily explaining a default, accompanied by payment of dues, the Council may, in its discretion, rescind any forfeiture of membership.

(*b*) *Misconduct:* The Council may, by a two-thirds vote of all the members thereof, censure, suspend or expel any member for misconduct in his relations to the Society, after written preferment of charges, thirty (30) days' written notice of hearing sent by registered mail, and an adequate opportunity to be heard before the Council or a committee of one or more MEMBERS designated by the Council.

Section 9. ***Effect of Termination of Membership and Lapse of Time.*** (*a*) All membership rights and interest in the property of the Society of persons resigning or otherwise ceasing to be members, or during the period of suspension of a suspended member, or on death, shall vest in the Society. (*b*) All acts of the Council which shall have received the express or implied sanction of the Society shall be deemed to be the acts of the Society and shall not thereafter be impeached by any member. (*c*) No claim, de-

mand, action or proceeding (individual, representative or derivative) shall be asserted or instituted by a member or former member against the Society, or any incumbent or former officer, Council or committee member thereof, by reason of service in any such capacities, after the expiration of six (6) months from the time when the same has accrued or actual or constructive knowledge has been acquired, or become available, of the facts upon which the same depends.

ARTICLE III

Chapters, Special and Student Branches, and Regional Areas

Section 1. ***Chapters.*** The Council may establish Chapters, which shall operate under the provisions of the Charter and the By-Laws of the Society, and the Rules and Regulations of the Council.

Section 2. ***Special Branches.*** Special Branches of the Society may be established, operated and maintained under the direction and in the discretion of the Council.

Section 3. ***Student Branches.*** Student Branches of the Society may be established, operated and maintained under the direction and in the discretion of the Council.

Section 4. ***Regional Areas.*** The Chapters and Branches of the Society in North America shall be divided by the Council into seven (7) geographical groups, and each group shall be designated a Regional Area. Delineation of Regional Areas, and changes therein, shall be published in an issue of the JOURNAL or mailed to all members.

Section 5. ***Chapter Membership.*** Chapters shall be composed of at least twenty (20) members of any and all grades, exclusive of Chapter Student members, residing in the vicinity of the Chapter's headquarters, and only members of the Society in good standing shall be eligible to become and remain Chapter members. Chapter members shall hold the same grade of membership in the Chapter as are held by them in the Society. No member shall vote or hold office concurrently in more than one (1) Chapter of the Society. All grades of Chapter members, except Students, shall be eligible to vote and hold office in Chapters.

Section 6. ***Limitations.*** The elected officers of Chapters shall receive no salary, emolument or compensation for their services as such. Chapters shall not act for the Society or subject the Society to any financial or other obligation, except such as the Society or the Council may by resolution assume. Notice to the foregoing effect shall be imprinted on the stationery used by each of the Chapters. Each Chapter shall promptly file a copy of its minutes with the Executive Secretary of the Society and make report to said Secretary of all of its proceedings. Chapters and Branches shall file with the chairmen of their respective Chapters Regional Committees their recommendations concerning the policies, procedures and operation of the Society, its Chapters and Branches. No contributions, except dues, shall be received or solicited by Chapters without the written approval of the Council. Chapters shall not issue publications or use the Society's name or emblem or Chapter insignia, without the approval of the Council. Chapters shall give no recommendations, endorsements or approvals of any scientific, literary, mechanical or engineering product for the promotion of private interests.

Section 7. ***Revocation of Charters.*** The charters of Chapters, Special Branches and Student Branches may be revoked by a two-thirds vote of all the members of the Council after written preferment of charges, sixty (60) days written notice of hearing sent by registered mail to the President of the Chapter or Branch, and an adequate opportunity to be heard before the Council or a committee of three (3) or more MEMBERS designated by the Council.

ARTICLE IV

Funds

Section 1. ***Admission Fees.*** Honorary Members and Students shall not be required to pay admission fees. The admission fees of Members, Associate Members and Affiliates shall be fixed by the Council, and shall be published in the JOURNAL. Admission fees shall accompany all applications for membership, and shall be refunded only in the event of the rejection of an application.

Section 2. ***Society Reserve Fund.*** Admission fees and such other funds as may from time to time be recommended by the Finance Committee and allocated by the Council shall be set aside and the principal thereof maintained as a Society Reserve Fund. Unless increased by the Society at an Annual or Special Meeting, the Society Reserve Fund shall be limited to a sum equal to fifteen dollars ($15.00) per member at the close of each fiscal year. The Council is hereby authorized and empowered, in any fiscal year in which the Society's revenues may be insufficient to meet expenses, to utilize a maximum of twenty percent (20%) of the Society Reserve Fund as valued on the first day of the fiscal year in which such a withdrawal may be required.

Section 3. ***Dues.*** Honorary Members, Presidential Members and Life Members shall be exempt from the payment of dues. The conferring of Fellow grade upon such members shall not affect such exemption. Unless changed by the Society at an Annual or Special Meeting, the annual dues of Fellows, Members, Associate Members thirty (30) years of age or over, and Affiliates shall be twenty-five dollars ($25.00). The annual dues of Associate Members under thirty (30) years of age shall be fifteen dollars ($15.00). The annual dues of Students shall be fixed by the Council and shall be published in the JOURNAL.

Annual dues, except Students, shall become due and payable in United States currency, or its equivalent, in advance on January 1 of each year, and shall in no case be subject to refund. Dues of new and advanced members, except Students, shall be prorated on a quarter-annual basis, and shall be payable on the first day of the month following notification of election or advancement, and if not paid within three (3) calendar months after such notification, the election or advancement shall be automatically rescinded. Students' dues shall be applicable to the annual period commencing October 1 and terminating September 30. The Council in its discretion may suspend payment of dues by members in the Armed Forces in time of war, or may in its discretion defer or remit the dues of any member for special cause. Future annual dues may be compounded at a three percent (3%) rate by payment to the Society of the worth of an annuity equal to the member's dues for the period for which dues would be required. Compounded payments shall in no event be subject to refund.

Section 4. ***Publications.*** Members of all grades, except Student and dues paying members whose pro-rated dues amount to less than ten dollars ($10.00), shall be entitled to receive the Society's TRANSACTIONS and GUIDE.

Section 5. ***Allocation of Dues for Research.*** Unless changed by the Society at an Annual or Special Meeting, forty percent (40%) of the dues received from Fellows, Members, Associate Members thirty (30) years of age or over, and Affiliates shall be allocated for basic or fundamental research in the principles and laws underlying matters in the arts relating to the sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

Section 6. ***Investment of Funds.*** (*a*) The Society Reserve Fund and such other funds as may be allocated by the Council for investment shall be invested and reinvested in bonds and obligations of the United States Government, the Government of the Dominion of Canada, or in investments legal for trust funds under the laws of the State of New York, subject to the proviso that not less than one-half of such Fund

and funds shall be invested in bonds and obligations of the United States Government; (*b*) gifts and bequests to the Society for a specific purpose or purposes shall, after acceptance by the Council, be used for the purposes specified and invested in the manner directed by the donor or testator, or in the absence of such direction, in the manner provided in subdivision (*a*) hereof; (*c*) the Council is authorized and empowered, in its discretion and without liability to the Society or to any member thereof, to retain any gift or bequest in property (real, personal or mixed) in the manner and form in which it shall be at the time of such gift or bequest.

Section 7. ***Budget.*** The Finance Committee shall submit to the Council at the Council's last quarter-annual meeting a Budget of estimated income and expenditures of the Society and all the committees thereof, for the succeeding fiscal year commencing November 1 and ending October 31. The expenditures of the Society's funds shall be governed by the Budget as approved, modified, or from time to time amended by the Council, and no additional expenditures shall be made without the approval of the Council.

Section 8. ***Receipts and Disbursements.*** The Executive Secretary shall render all bills and collect all moneys due the Society, and shall enter all receipts in the Society's books and deposit the same to the Treasurer's Account in banks designated by the Council. Except as hereinafter provided, no contract or other obligation for the payment of money shall be valid unless signed or countersigned by the Executive Secretary. Except as hereinafter provided, all vouchers against the Society for the payment of funds shall be submitted to the Executive Secretary, who shall certify the correctness thereof and the authorization thereof by the Budget, and payment shall be made only upon such certification.

Section 9. ***Bonds.*** The Treasurer, the Executive Secretary, and all other Officers, Agents or Employees authorized by the Council to endorse or execute drafts for the payment of money, shall give bond in a penal sum and with sureties approved by the Council, for the faithful performance of their duties, the premiums therefor, if any, to be paid by the Society.

Section 10. ***Audits.*** Between the close of the fiscal year and January 1 of each year, the accounts of the Society shall be audited by a certified public accountant approved by the Council, and the auditor's report shall be presented by the Treasurer at the Annual Meeting of the Society, and shall be published in the JOURNAL.

ARTICLE V

The Council

Section 1. ***Members.*** The Council shall consist of the last living Past President, the President, the First Vice President, the Second Vice President, the Treasurer and twelve (12) MEMBERS, seven (7) of whom shall be from different Regional Areas and elected as Regional Directors for their respective Areas. The twelve (12) MEMBERS shall be divided into three (3) classes of four (4) in each class, and the MEMBERS in each class shall hold office for three (3) years and until their successors shall have been elected and installed. Four (4) MEMBERS shall be elected to the Council at each Annual Meeting, and also such additional number, if any, as may be necessary to fill vacancies. Pending the Annual Meeting, vacancies occurring in the Council may be filled by the Council. The Executive Secretary of the Society shall be the Secretary of the Council. He may take part in the deliberations of the Council but shall not be a member thereof or have any vote therein.

Section 2. ***Powers.*** In addition to the powers specifically conferred upon the Council (directors) by the laws of the State of New York, the Charter and these By-Laws, the

Council shall have the general management and full control of the affairs and all activities of the Society and, subject to the Charter, the By-Laws and the laws of the State of New York, may, in its discretion, promulgate and amend Rules and Regulations, and issue directives for the administration of the Society's affairs and the regulation of all Committees, Chapters, Branches, Regional Areas, Officers, Agents and Employees. The Council may, in its discretion, refer to the Society any important question pertaining to the Society, and shall refer any such question to the Society upon a majority vote taken at a stated or Special Meeting held by the Society.

Section 3. ***Meeting, Quorum and Reports.*** The Council shall hold its annual meeting as soon as practicable after the close of the Annual Meeting of the Society, and shall hold meetings quarter-annually thereafter. Special meetings may be called by the President or by three (3) Council members. A majority of the Council members in office shall constitute a quorum. The Council shall keep a record of its proceedings, and shall report on its activities at each meeting of the Society and, at the Annual Meeting, it shall present a written report as required by the Membership Corporations Law of the State of New York.

Section 4. ***Notice of Council Meetings.*** Unless waived in writing or by telegraph or cable, notice of any quarterly or special meeting of the Council shall be given in writing, mailed to the last known address of each Council member, by the Executive Secretary or the President, or the three (3) Council members calling the meeting, not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting.

ARTICLE VI

Officers

Section 1. ***Elected Officers.*** The elected officers of the Society shall be a President, a First Vice President, a Second Vice President and a Treasurer. The elected officers shall receive no salary, emolument or compensation for services rendered to the Society, and shall serve for one (1) year and until their respective successors shall be elected and installed.

Section 2. ***Appointed Officers.*** The Executive Secretary shall be appointed by the Council at its annual meeting, at a salary fixed by the Council, to serve for one (1) year and until his successor shall be appointed. The Executive Secretary shall be subject to removal by a two-thirds vote of the Council cast by secret ballot.

Section 3. ***Presiding Officer.*** At all meetings of the Society and of the Council, the President, or in his absence, the Vice Presidents in order of seniority, or in their absence the Treasurer or a MEMBER selected by the Council, shall preside.

Section 4. ***The President.*** The President shall exercise the powers and duties assigned to him by these By-Laws and, subject to the direction of the Council, he shall be the chief executive officer of the Society and generally supervise its affairs. At the Annual Meeting of the Society he shall make a report relative to the Society's condition, activities and progress. No President may be re-elected to that office until the Annual Meeting next following the expiration of his term.

Section 5. ***Vice Presidents.*** In the absence, disability, resignation or death of the President, the Vice Presidents in order of seniority, shall exercise the powers and perform the duties of the President.

Section 6. ***The Treasurer.*** The Treasurer shall have custody of the funds of the Society and the Society's books of account, which shall be open to the inspection of any member of the Council. Pending the Annual Meeting, a vacancy occurring in the office of Treasurer shall be filled by the Council.

Section 7. ***The Executive Secretary.*** The Executive Secretary shall be the chief administrative officer of the Society and the manager of the Society's publications. Copies of the minutes of all meetings of the Society, of the Council, and of all committees shall be filed with him. He shall keep such books, papers and records as the Society or the Council may direct, which shall be open to the inspection of any member of the Society. He shall be the keeper of the seal of the Society, and may in his discretion use the designation "Secretary" on legal documents. He shall conduct the correspondence of the Society and shall keep full records of the same. He shall promptly notify the members of the Council, the officers, the nominated candidates, the members of all committees, and applicants for admission or advancement, of their election, nomination, appointment or advancement. He shall issue notices of all meetings of the Society, and, in the case of Special Meetings, he shall add a brief note of the object of the call. He shall be in charge of the offices of the Society and shall administer them under such rules of procedure as the Council may approve. Subject to the discretion of the Council, the Executive Secretary may employ such assistant secretaries and other personnel as may be deemed to be necessary.

Section 8. ***Other Duties.*** All officers of the Society shall perform the duties customarily attaching to their respective offices under the laws of the State of New York, and such other duties and services incident to their respective offices as are delegated to them in these By-Laws and as may from time to time be assigned to them by the Council.

ARTICLE VII

Committees

Section 1. ***Advisory Board.*** The Advisory Board shall consist of all Presidential Members, of which the last living Past President shall be chairman, and the said committee shall consider and make recommendations to the Council concerning matters of policy affecting the Society referred to the Board by the Council.

Section 2. ***Council Committees.*** Unless otherwise provided, the Council Committees and the respective chairmen thereof shall be appointed by the President, with the approval of the Council, as soon as practical after the close of the Annual Meeting. Each Council Committee, except Regions Central Committee, shall consist of not more than five (5) members, of whom not less than three (3) shall be Council members. Members of the Committees shall serve for a term of one (1) year and until their successors are chosen. The following shall be the Council Committees and their respective duties:

(*a*) *Executive Committee*, which shall investigate and make reports and recommendations to the Council regarding matters relating to the Society or any member or members thereof. During intervals between Council meetings the Executive Committee may exercise such powers of the Council as may lawfully be delegated to it by the Council.

(*b*) *Finance Committee*, which under the direction of the Council shall supervise and control the financial affairs of the Society and its books of account. It shall survey, investigate and analyze all financial requirements and expenditures, scrutinize all Budget estimates, and prepare the Budget for submission to the Council. The Treasurer shall be an ex-officio member of the said committee, with the power to vote.

(*c*) *Membership Committee*, which shall publicize the aims, activities, achievements and the scientific and educational purposes of the Society, toward the end that persons duly qualified shall apply for membership therein.

(*d*) *Program and Papers Committee*, which shall plan the general character of all technical meetings of the Society, solicit and receive papers for consideration by the

Publication Committee, and select from the papers which have been approved by the Publication Committee those for presentation at technical sessions.

(*e*) *Regions Central Committee*, consisting of the Second Vice President and the seven (7) Regional Directors. The said committee shall consider and report to the Council on the activities of Chapters and Branches, and make recommendations to the Council concerning the policies, procedures and operation of the Society, its Chapters and Branches; it shall coordinate the activities of the Chapters Regional Committees; and it shall investigate applications for the creation of Chapters and Branches, and report thereon to the Council. The Second Vice President shall be the chairman of said committee.

Section 3. **General Committees.** Unless otherwise provided, the General Committees, and the respective Chairmen thereof, shall be appointed by the President, with the approval of the Council, as soon as practicable after the close of the Annual Meeting. The President, with the approval of the Council, shall at the same time appoint the chairman and the vice chairman of the Committee on Research. The chairman of said committee shall have had at least one (1) year of service on said committee. The vice chairman shall perform the duties of the chairman in the latter's absence, disability, resignation or death. The following shall be the General Committees and their respective duties:

(*a*) *Admission and Advancement Committee*, consisting of three (3) MEMBERS in good standing as MEMBERS for at least ten (10) years and having at least three (3) years of service on the Council or on General Committees, which shall receive and consider applications for admission or advancement to Member, Associate Member, or Affiliate grade, make diligent scrutiny and inquiry as to the character and qualifications of applicants, and report to the Council on the eligibility of each applicant for admission or advancement. The correspondence and proceedings of said committee shall be secret and confidential, and its correspondence concerning unsuccessful applicants shall be destroyed within a reasonable time.

(*b*) *Publication Committee*, consisting of three (3) MEMBERS. Subject to the direction of the Council the said committee shall formulate the editorial policies of the Society and for all of its publications. The chairman of the said committee may appoint sub-committees of one (1) or more members to review and report to the committee on the quality and appropriateness for publication of papers and bulletins intended for presentation or presented at Society meetings and the discussions thereof. In the performance of its functions the said committee and its sub-committees shall be subject to the following conditions: (*a*) That the data recommended for publication shall tend toward the professional education of the individual engineer; (*b*) that such data shall be free from commercial bias; and (*c*) that such data shall tend to advance for the public benefit the sciences relating to the arts of heating, ventilating, cooling or air conditioning, or the allied arts and sciences.

(*c*) *Guide Committee*, consisting of nine (9) MEMBERS, to serve for a term of one (1) year commencing November 1. The said committee, in harmony with the editorial policies of the Society, shall compile the text section of THE GUIDE. The chairman of the Committee on Research shall be an ex-officio member of the said committee.

(*d*) *Charter and By-Laws Committee*, consisting of three (3) MEMBERS, which shall consider all matters affecting the Charter and By-Laws, Rules and Regulations, and make recommendations thereon to the Council.

(*e*) *F. Paul Anderson Committee*, consisting of the First Vice President and four (4) MEMBERS, to serve for a term of one (1) year, which shall receive from the Council the announced conditions for the making of the subsequent annual award of the F. Paul Anderson Medal, select, solicit and carefully consider candidates for the award, and make nominations to the Council of the members complying with the announced conditions and deemed worthy to be recipients of the award. The decision to confer the award and all matters relating thereto shall be by a majority vote of all the members of the Council and in accordance with the presentation letter of Thornton Lewis.

(*f*) *Chapters Regional Committees*, each serving one Regional Area, and each consisting of the Regional Director for the Area and one (1) member and one (1) alternate member selected by each Chapter therein, to serve for a term of one (1) year. The said committees shall solicit from the Chapters and Branches within their respective Regional Areas recommendations concerning the policies, procedures, and operation of the Society, its Chapters and Branches, review the same, and make recommendations thereon to the Regions Central Committee. Said committees shall select the MEMBERS and alternates to serve on the Nominating Committee, and duly notify the Executive Secretary of such selections. The alternate members of Chapters Regional Committees may be present at committee meetings and participate in the deliberations thereof, but shall not vote therein except in the absence of the committee members for whom they respectively are alternates. The said committees shall hold committee meetings prior to June 1 of each year. Each Regional Director shall be the chairman of the Committee serving his Regional Area.

(*h*) *Committee on Research*, consisting of fifteen (15) MEMBERS, nominated by the Council or as provided in **ARTICLE VIII,** *Section 4*, and elected by the Society in the manner of elected officers. Subject to the direction of the Council, the said committee shall conduct and coordinate fundamental and basic research, and study and determine the principles and laws underlying matters in the sciences relating to the arts of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, and cooperate with universities, colleges, schools and other organizations and groups, including governmental agencies, in the investigation of research subjects, subject to the proviso that all of the foregoing activities shall be devoted to the public welfare and general benefit, and shall not be designed to promote any individual, private or commercial interests. No contributor shall be specially favored on account of any contribution for research, which shall be used only for the public welfare. The said committee shall submit to the Publication Committee and to the Executive Secretary of the Society, its papers and reports concerning its investigations and activities.

There shall be a *Research Executive Committee*, consisting of the chairman, the vice chairman and three (3) other members of the Committee on Research appointed by the chairman, which shall exercise the powers and carry out the purpose of the Committee on Research during intervals between Committee meetings.

The chairman of the Committee on Research shall appoint such *Technical Advisory Committees* as may be deemed expedient to advise the Committee on Research and the Director of Research on specific research projects. At least one (1) member of each Technical Advisory Committee shall be a member of the Committee on Research and the chairman of the Committee on Research and the Director of Research shall be ex-officio members thereof. Technical Advisory Committees shall be governed by such rules and regulations as may be recommended by the Committee on Research and adopted by the Council.

The chairman of the Committee on Research may appoint a *Technical Adviser* as the Committee's consultant, who shall serve without compensation.

The Committee on Research shall recommend to the Council for appointment a *Director of Research*, whose activities, proceedings and reports shall be subject to the direction of the committee and the approval of the Council. The salary of the Director of Research shall be fixed by the Council upon the recommendation of the Committee on Research, and his employment may be terminated in the Council's discretion.

The Director of Research shall direct the operations of the Society's research program. Subject to the approval of the Committee on Research, the Director of Research may employ such assistants and other personnel as may be deemed by said committee to be necessary. The Director of Research shall be provided with a revolving fund in an amount fixed by the Council upon the recommendation of the Finance Committee, for the payment of the compensation of part time or temporary employees, travelling expenses and incidental petty cash outlays. He shall approve all purchase invoices and disbursements for research, and submit vouchers for the payment thereof at least semi-

monthly to the Executive Secretary. The Director of Research shall assist the Committee on Research in the preparation of its Budget for submission to the Finance Committee of the Society. The Director of Research and such other research assistants as may be employed in connection with research, shall, in consideration of such employment, agree in writing that any inventions, discoveries, ideas, plans, processes, formulae, experimental results and information received, published, divulged, made, or developed, or which may be in the course thereof while engaged in such employment, shall belong to the Society, and that no material relating thereto will be submitted elsewhere without the Council's consent.

(*i*) *Public Relations Committee*, consisting of a member of Council and three (3) members who shall serve a term of one (1) year. The said committee shall publicize the aims, activities and achievements, the scientific, and educational purposes of the Society, with the object of cultivating and stimulating public and members' interest in the Society and its affairs.

(*j*) *Standards Committee*, consisting of six (6) Members. The said Committee shall consider all scientific questions and data pertaining to engineering codes and standards, initiate and propose changes and improvements thereof for the public benefit, and report its recommendations to the Council.

Section 4. ***Other Committees.*** The Council may from time to time appoint other committees of one (1) or more members, and define their powers and duties, and it may abolish any such committees.

Section 5. ***General Provisions Concerning Committees.*** All Council, General and Other Committees of the Society, except the Nominating Committee, are subject to the following provisions:

(*a*) Except as otherwise provided, members on General Committees shall be divided into three (3) classes of equal number, each class to serve for three (3) years, and one-third of the prescribed number to be appointed, or in the case of the Committee on Research elected, annually.

(*b*) Except as otherwise provided, the committee year of all committees shall run from the close of the Annual Meeting to the close of the next Annual Meeting. The committee members whose terms expire shall continue in office until their respective successors are appointed or elected.

(*c*) Except as otherwise provided, each committee shall meet as soon as practicable after its appointment or election and may hold semi-annual meetings and such special meetings as the chairman or one-third of the members thereof may call. Each committee may recommend rules of procedure, which shall become operative upon the approval of the Council. Unless waived in writing or by telegraph or cable, notice of all semi-annual and special meetings of all committees shall be given in writing to all committee members and to the Executive Secretary not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting. At committee meetings a majority of the members in office shall constitute a quorum. The committees shall promptly and fully report their activities to the Council and file with the Executive Secretary the minutes of their meetings and a complete record of their proceedings.

(*d*) The Council may by a two-thirds vote remove a member of any committee, and may, by a majority vote, designate a member to fill the vacancy or any other vacancy arising.

(*e*) Each committee shall submit to the Finance Committee on or before September 15 of each year an estimate of its disbursements for the ensuing fiscal year, and no committee, or member thereof, shall have the authority to incur any indebtedness or pecuniary obligation for which the Society shall be responsible, or claim reimbursement for advances, except to the extent authorized in the Budget or by resolution of the Council. Each committee member shall be responsible for the proper application of all funds remitted to him.

(*f*) Each committee's actions, proceedings, findings, conclusions and reports shall be subject to the direction and review of the Council, and the Council may take such steps, or see that such steps are taken by the committees as may be appropriate to comply with the Charter and By-Laws, and to make effective any resolution adopted by the Society or any resolution, rule or directive of the Council.

(*g*) If any doubt or controversy should arise as to whether a particular subject or matter is within the jurisdiction of a committee or whether any action should be taken by a committee, or in the case of a committee tie vote, the same shall be settled and determined by the Council.

(*h*) Except as otherwise provided, ex-officio members of committees may participate in committee deliberations but shall have no vote therein.

ARTICLE VIII

Meetings, Nominations and Elections

Section 1. ***Meetings.*** The Annual Meeting of the Society shall commence during the thirty (30)-day period beginning with the fourth Monday in January, and shall continue from day to day as the Council may arrange. Semi-Annual Meetings shall be held at such times as may be fixed by the Council. Special Meetings may be called at any time by the Council, and shall be called by the Council upon the written request of the President or of fifty (50) MEMBERS of the Society. Meetings shall be held at such place or places as the Council may designate, and shall be governed by Robert's Rules of Order, Revised, except when inconsistent with the laws of the State of New York, the Charter or these By-Laws. At any meeting of the Society, the presence of fifty (50) members entitled to vote shall be necessary to constitute a quorum.

Section 2. ***Notices.*** Notice of the Annual, Semi-Annual and of any Special Meeting of the Society shall be given in writing by the Executive Secretary and mailed, postage prepaid, not less than twenty (20) nor more than forty (40) days before the date fixed for the meeting, to each member of the Society at his last known address appearing on the records of the Society, and shall be published in the JOURNAL. Notices of Special Meetings shall state the purpose or purposes for which the meeting is called, and no business other than that set forth in the notice shall be entertained or transacted thereat.

Section 3. ***Nominating Committee.*** The Nominating Committee, to serve for a term of one (1) year from the opening of the Annual Meeting, shall consist of eleven (11) MEMBERS and sixteen (16) alternate MEMBERS, all of whom shall have been in good standing as MEMBERS for a period of at least five (5) years. Four (4) members and one (1) first alternate and one (1) second alternate member of the said committee, each from a different Regional Area, shall be selected by the Council at or prior to its last quarter-annual meeting; and one (1) member and one (1) first and one (1) second alternate member shall be selected by each Chapters Regional Committee, and certified to the Executive Secretary by September 1 of each year. No member of the Council shall be eligible to serve on the Nominating Committee, and no Chapter shall be represented on the said committee by more than one (1) of the eleven (11) MEMBERS. The Nominating Committee so chosen shall effect its own organization during the Annual Meeting of the Society, and shall hold a meeting during the Semi-Annual Meeting of the Society. By September 1 of each year the Nominating Committee shall nominate candidates for the elective offices, the members of the Council and the Regional Directors to be elected at the ensuing Annual Meeting, and notify the Executive Secretary of the names of the nominees, the notice to be accompanied by the written acceptances of the candidates. Seven (7) members entitled to vote on the said committee shall constitute a quorum. No alternate member may be present at committee meetings, nor shall he participate in the deliberations thereof or vote therein except in

the absence of the member, or the member and the first alternate member of the one of the eight (8) groups with which he was selected. The transportation expenses, as defined by the Council, of eleven (11) committee members who participate in the deliberations at the annual and semi-annual meetings of said committee shall be included in the budget and defrayed by the Society.

Section 4. ***Other Nominations.*** Nominations of officers and members of the Council, other than those nominated by the Nominating Committee, and nominations of members of the Committee on Research, other than those nominated by the Council, may be made in writing by at least fifty (50) members eligible to vote, upon presentation of such nominations, with each nominee's consent, to the Executive Secretary at least sixty (60) days prior to the opening of the first session of the Annual Meeting, whereupon the nominees' names shall be placed upon the ballot with a notation that they are presented by members independent of the Nominating Committee.

Section 5. ***Voting.*** Voting at any meeting may be in person or by proxy, but only the Executive Secretary and MEMBERS of the Society shall be eligible to act as proxies. Proxies shall not be valid for more than three (3) months from dates of execution. The Executive Secretary and the MEMBERS acting as proxies shall hold the ballots of their principals secret and confidential. Voting for election of officers, Council members, members of the Committee on Research, on proposals to amend these By-Laws, and on questions required to be referred to the Society pursuant to **ARTICLE V**, *Section 2*, shall be by secret ballot. In the event of any tie vote, the Council shall decide the vote.

Section 6. ***Ballots.*** Together with notice of the Annual Meeting, the Executive Secretary shall forward appropriate proxies and ballots to members entitled to vote. The proxies and ballots shall contain spaces for write-in names.

Section 7. ***Results.*** The polls for election shall be opened at the opening of the Annual Meeting and shall remain open for a period of five (5) hours. Thereafter the ballots shall be opened by three (3) inspectors of election appointed by the President, who shall be authorized to fill any vacancy occurring among such inspectors. The inspectors of election shall consider ballots and votes to be valid provided the intent of the voter is clear. The result of the vote shall be reported by the inspectors of election in writing, and shall be announced by the President on the second day of the Annual Meeting, whereupon the terms of the inspectors of election shall expire. The elected candidates shall be installed during the Annual Meeting and their terms shall commence at the close of the last session of the Annual Meeting.

ARTICLE IX

Amendments

Section 1. ***Prerequisites.*** These By-Laws may be amended by a two-thirds vote of the Society at an Annual Meeting thereof, provided that written notice of the proposed amendment, subscribed by two-thirds of the members of the Council or by fifty (50) MEMBERS, be given at a previous stated or Special Meeting, and that notice thereof as pertinently amended by majority vote at said stated or Special Meeting be also given by the Executive Secretary in the notice of the Annual Meeting.

Section 2. ***Renumbering.*** The Council may, by a two-thirds vote, renumber existing Articles or Sections of these By-Laws.

ARTICLE X

Adoption

Section 1. ***Effect.*** These By-Laws shall supersede all previous Constitutions, By-Laws and Rules of the Society, and shall come into effect upon the adjournment of the meeting at which these By-Laws shall be adopted.

1622

No. 1622

DIRECT SOLAR RADIATION AVAILABLE ON CLEAR DAYS†

By J. L. Threlkeld* and R. C. Jordan**, Minneapolis, Minn.

ENGINEERING and scientific problems involved with the irradiation of surfaces on the earth by the sun are continually increasing. In this country and abroad, research on fundamental and applied topics on harnessing of solar energy in thermal, electrical, and chemical processes is expanding. In the field of thermal utilization of solar energy, some applications are already of commercial practicability. The solar furnace[1] is becoming an important research tool in the production of controlled high temperatures, while in the realm of low temperature utilization, solar energy has been put to commercial use in the space heating of buildings.[2]

Of equal interest and at the present time of greater economic importance is the minimizing or prevention of entry or incidence of solar radiation. In the expanding field of summer air conditioning, solar radiation may represent a sizable portion of the cooling load. Shading devices of various types and reflecting surfaces are used to reduce the heating effect of the sun's rays.

In all types of solar radiation problems, regardless of whether they involve utilization or control, quantitative knowledge of incident solar energy is needed. Solar energy availability knowledge is needed for all types of solar weather including clear, partly cloudy, and cloudy days.

Clear-day knowledge of solar availability is of much importance. In the field of solar energy control, design problems involve clear days when solar effects are large. In solar energy utilization problems, apparatus is often designed for clear-day operation. In addition, a study of clear-day radiation is a logical start into the problem of incidence of solar energy on partly cloudy and cloudy days.

In most solar energy problems, knowledge of both direct and diffuse radiation is needed. In the case of concentrating devices, such as a solar furnace, only direct or beamed radiation is utilized. During clear days the diffuse component is usually small when compared to the direct component.

---

†This research was sponsored by the *National Science Foundation*.
*Professor, Department of Mechanical Engineering, University of Minnesota. Member of ASHAE.
**Professor and Head, Department of Mechanical Engineering, University of Minnesota. Member of ASHAE.
[1]Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, Pa., January 1958.

This paper is concerned only with the incidence of direct solar radiation during clear days. An associated paper on diffuse radiation is planned for future publication.

FUNDAMENTAL PROCEDURE

When the earth is at its mean distance from the sun, the solar radiation incident upon a surface normal to the sun's rays and at the outer limit of the earth's atmosphere is known as the solar constant. Johnson[3] has determined the solar constant to be 2.00 g-cal per (sq cm) (min) or 442.4 Btu per (hr) (sq ft) with a probable error of ± 2 percent.

As the solar beam passes through the atmosphere, it is split into 3 parts: (1) one part is turned aside notably by air molecules, water vapor molecules, and dust particles and scattered in all directions; (2) a second part is absorbed, chiefly by water vapor and ozone; and (3) the remainder passes through the atmosphere unchanged in wavelength.

The mechanisms of scattering and absorption by atmospheric constituents have been studied by many investigators. In a recent paper, Fritz[4] summarized the available knowledge and presented recommended procedures for calculating these effects.

The fundamental method used here is largely an extension of Moon's[5] procedure using the best known available data. Moon correlated the work of several investigators on the spectral total effect of scattering by

$$\tau_s = [(\tau_a)^{p/760}(\tau_w)^{w/20}(\tau_d)^{d/800}]^m \quad . . . . \quad (1)$$

Moon also calculated transmissivities due to absorption by ozone and water vapor. By combining these transmissivities with Equation 1, the total spectral transmissivity is given by

$$\tau = (\tau_a)^{mp/760}(\tau_w)^{mw/20}(\tau_d)^{md/800}(\tau_o)^{mo/2.5}(\tau_{wA})^{mw/20} \quad . . . . . . \quad (2)$$

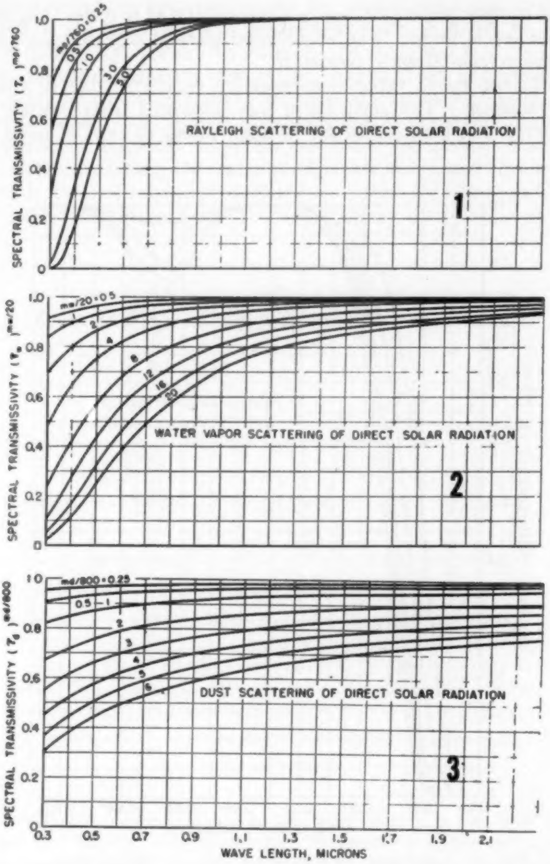
The exponential constants of Equation 2 represent in each case the magnitude for which the respective individual transmissivity is known for unit air mass.

Fig. 1 shows the dry air scattering transmissivity $(\tau_a)^{mp/760}$ as a function of wavelength and the exponent mp/760. The curve for mp/760 equal to unity was presented by Fritz.[4] The other curves were calculated from the unity curve. It is significant to observe that the depletion effect of dry air or Rayleigh scattering is primarily at the shorter wavelengths and at high air masses it becomes pronounced.

Fig. 2 shows the water vapor scattering transmissivity $(\tau_w)^{mw/20}$ as a function of wavelength and the exponent mw/20. The unity curve for unit air mass and 20 mm of precipitable water was taken from Moon's[5] paper. Fig. 2 shows that for large values of the exponent mw/20, water vapor significantly reduces the direct solar beam by scattering. As compared to dry air scattering, water vapor scattering extends to much higher wavelengths.

Fig. 3 shows the dust scattering transmissivity $(\tau_d)^{md/800}$ as a function of wavelength and the exponent md/800. The unity curve for unit mass and 800 dust particles per cc was taken from Moon's[5] paper. Fig. 3 shows that dust scattering is less influenced by wavelength than are dry air and water vapor scattering.

Fig. 4 shows the ozone transmissivity $(\tau_o)^{mo/2.5}$ due to absorption of direct solar radiation as a function of wavelength and the exponent mo/2.5. The unity curve for unit air mass and 2.5 mm of ozone depth was calculated from Moon's



FIGS. 1, 2, 3—SPECTRAL FRACTION OF DIRECT SOLAR RADIATION TRANSMITTED THROUGH ATMOSPHERE DUE TO SCATTERING; 1 IS RAYLEIGH SCATTERING AT VARIOUS BAROMETRIC PRESSURES AND AIR MASSES; 2 IS SCATTERING AT VARIOUS DEPTHS OF PRECIPITABLE WATER AND AIR MASSES; 3 IS DUST SCATTERING AT VARIOUS CONCENTRATIONS OF DUST PARTICLES AND AIR MASSES

data[4]. Fritz[4] has stated that 2.5 mm of ozone (NTP) is typical of middle latitudes and is concentrated mainly at elevations between 15 and 35 kilometers. Fig. 4 shows that ozone absorbs selectively, being strongly absorbing in the ultraviolet

band and also depletes some radiation in the band of 0.48 to 0.70μ. In other bands the transmissivity is unity. The exponential method of extension to other air masses was taken from Moon's[5] procedure.

Fig. 5 shows the water vapor transmissivity $(\tau_{wA})^{mw/20}$ due to absorption of direct solar radiation. The unity curve was calculated from Fowle's[6] data. A family of curves was constructed for various values of mw/20 using the same

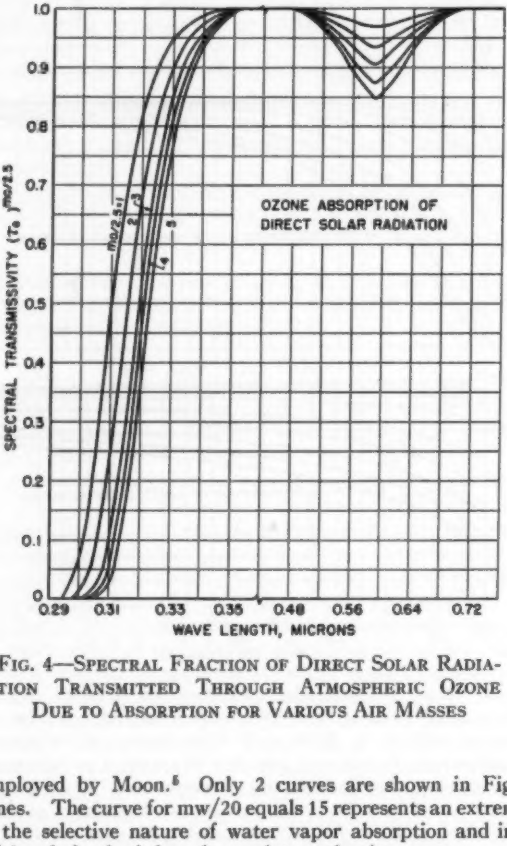


FIG. 4—SPECTRAL FRACTION OF DIRECT SOLAR RADIATION TRANSMITTED THROUGH ATMOSPHERIC OZONE DUE TO ABSORPTION FOR VARIOUS AIR MASSES

method as employed by Moon.[5] Only 2 curves are shown in Fig. 5 to avoid confusion of lines. The curve for mw/20 equals 15 represents an extreme condition. Fig. 5 shows the selective nature of water vapor absorption and indicates that in some broad bands in the infrared complete extinction occurs.

With the fundamental data given by Figs. 1 through 5, total spectral transmissivities, τ, may be calculated by Equation 2 for various air masses and atmospheric conditions. At any wavelength, the total spectral transmissivity multiplied by the spectral radiation intensity outside the atmosphere gives the spectral direct radiation intensity at the earth's surface.

Complete calculations were carried out for the following conditions: Sea level; Ozone depth = 2.5 mm; Depth of precipitable water = 0, 10, 30, 60 mm; Dust particles = 0, 400, 800, per cc; Air mass = 1, 2, 3, 4, and 5.

In all, 60 sets of calculations were made. For each set, individual calculations for τ were made for 63 wavelengths in the range of 0.295 to 2.13 μ, thus giving sufficient points to construct accurately the energy curves at the earth's surface.

Fig. 6 shows the graphical results for one set of calculations where the conditions were 400 dust particles, 30 mm of precipitable water, and air mass of 1. The upper curve is the spectral energy distribution outside the atmosphere as given by Johnson.[3] The lower curve is the calculated energy distribution at sea level for

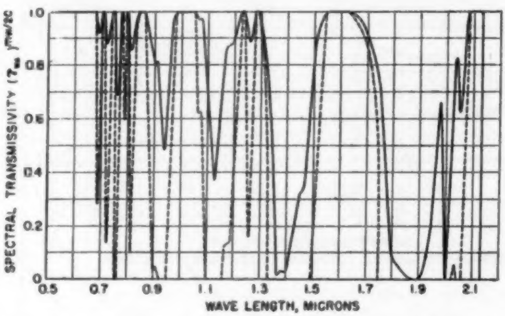


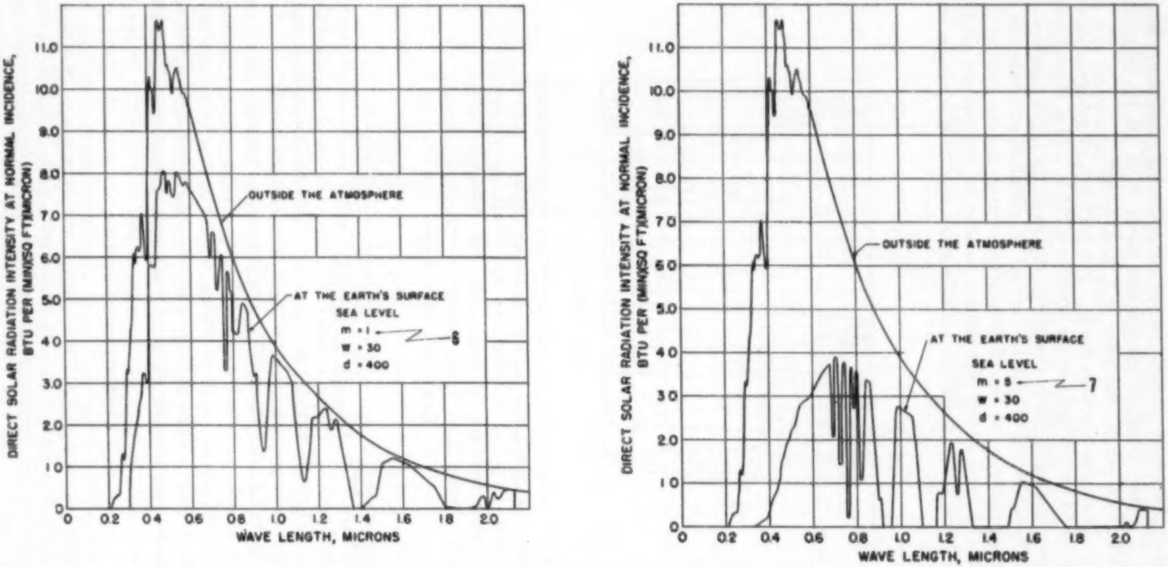
FIG. 5—SPECTRAL FRACTION OF SOLAR DIRECT RADIATION TRANSMITTED THROUGH ATMOSPHERIC WATER VAPOR DUE TO ABSORPTION FOR VARIOUS DEPTHS OF PRECIPITABLE WATER AND AIR MASSES; WITH SOLID LINES INDICATING MW/20 = 1.0; AND BROKEN LINES MW/20 = 15.0

the given conditions. The area under the upper curve is the solar constant or 442.4 Btu per (hr) (sq ft). The area under the lower curve as determined through use of a planimeter is 280.0 Btu per (hr) (sq ft). Thus the atmospheric transmission factor for the assumed conditions is 280.0/442.4 equals 0.633.

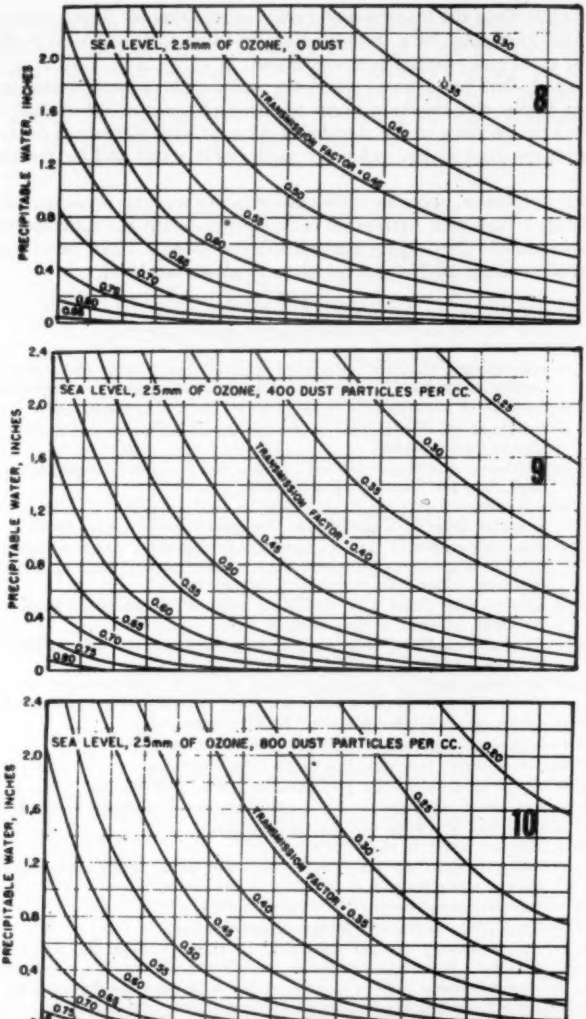
Fig. 7 shows the results for the same conditions as Fig. 6 except for air mass 5. For this case the transmission factor is 0.276. Fig. 6 represents the condition for the sun at the zenith (altitude angle of 90 deg) while Fig. 7 is for an altitude angle of approximately 11 deg. Thus the length of path of the sun's rays through the atmosphere is of extreme importance in affecting reduction of solar intensity.

TRANSMISSION FACTORS

The calculations previously described allowed the calculation of atmospheric transmission factors for a wide variety of conditions. Figs. 8, 9, and 10 show the transmission factor for the atmosphere as a function of precipitable water and air mass for 0, 400, and 800 dust particles per cc, respectively. These figures show



FIGS. 6, 7—SPECTRAL DISTRIBUTION OF DIRECT SOLAR RADIATION AT NORMAL INCIDENCE FOR THE UPPER LIMIT OF THE ATMOSPHERE AND AT THE EARTH'S SURFACE FOR CONDITIONS AT SEA LEVEL, 30 MM OF PRECIPITABLE WATER, AND 400 DUST PARTICLES PER CC; M = 1 AT 6 AND M = 5 AT 7



Figs. 8, 9, 10—Atmospheric Transmission Factor for Direct Solar Radiation at Normal Incidence for Sea Level, 2.5 mm of Ozone, and Various Dust Particles as a Function of Precipitable Water and Air Mass; with Zero Dust at 8; 400 Dust Particles per cc at 9; 800 Dust Particles per cc at 10

that during clear days the transmission factor at sea level for direct solar radiation at normal incidence may vary from about 0.85 down to less than 0.20 depending upon the air mass and atmospheric conditions.

It was found that for a given air mass and depth of precipitable water the transmission factor was closely a linear function of the number of dust particles. Fig. 11 shows the decrease of the transmission factor with increase of dust, with the decrease being greatest at low values of precipitable water and large values of air mass.

The effect of elevation upon the transmission factor was also investigated. Equation 2 and Fig. 1 show that with other conditions constant the total spectral transmissivity increases as the barometric pressure decreases. Thus, one would expect the transmission factor to increase for surfaces above sea level.

NOMENCLATURE

$C.N.$ = atmospheric clearness number, ratio of direct solar radiation at normal incidence for locality to that for basic atmosphere, dimensionless.
d = number of atmospheric dust particles per cubic centimeter.
I_H = incidence of direct solar radiation upon a horizontal surface, Btu per (hour) (square foot).
I_N = incidence of direct solar radiation upon a surface perpendicular to sun's rays, Btu per (hour) (square foot).
I_o = incidence of solar radiation upon a surface perpendicular to sun's rays at outer limit of atmosphere, Btu per (hour) (square foot).
I_{sc} = the solar constant; incidence of solar radiation upon a surface perpendicular to sun's rays at outer limit of atmosphere and at the mean solar distance, 442.4 Btu per (hour) (square foot).
I_{TV} = incidence of direct solar radiation upon a surface tilted from the vertical position, Btu per (hour) (square foot).
I_v = incidence of direct solar radiation upon a vertical suface, Btu per (hour) (square foot).
m = air mass, ratio of length of path of sun's rays through atmosphere to vertical length of path when sun is in zenith position, dimensionless.
o = depth of atmospheric ozone (NTP), millimeters.
p = barometric pressure, millimeters of mercury.
TF = atmospheric transmission factor, dimensionless.
w = depth of atmospheric precipitable water, millimeters.
α = wall-solar azimuth angle; angle between horizontal projection of sun's rays and perpendicular to vertical wall.
β = altitude angle of sun.
γ = azimuth angle of sun; angle measured from north to horizontal projection of sun's rays.
μ = wavelength, microns.
τ = total spectral transmissivity, dimensionless.
τ_a = transmissivity due to scattering by dry air molecules for unit air mass and for a barometric pressure of 760 mm of mercury, dimensionless.
τ_d = transmissivity due to scattering by dust particles for unit air mass and for 800 dust particles per cubic centimeter, dimensionless.
τ_o = transmissivity due to absorption by ozone for unit air mass and for 2.5 mm of ozone depth, dimensionless.
τ_s = total spectral transmissivity due to scattering, dimensionless.
τ_w = transmissivity due to scattering by water vapor molecules for unit air mass and for 20 mm of precipitable water, dimensionless.
τ_{wA} = transmissivity due to absorption by water vapor for unit air mass and for 20 mm of precipitable water, dimensionless.
Φ = angle of tilt from vertical position.

A set of calculations were made for 1000 ft elevation, 2.5 mm of ozone, 30 mm of precipitable water, 400 dust particles per cc, and for 5 air mass values. Transmission factors were determined and were found to differ from the sea level transmission factor for the same condition by about one percent. Thus for ordinary purposes the sea level transmission factors as given by Figs. 8 through 11 are sufficiently accurate at elevations up to 1000 ft.

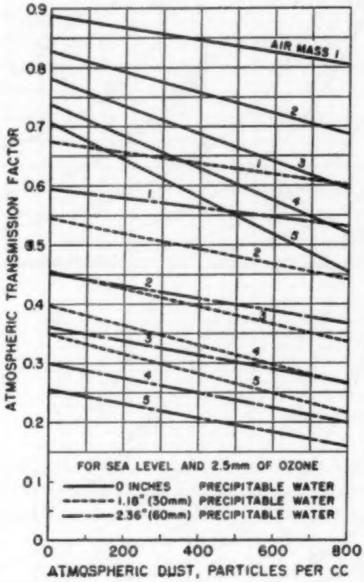
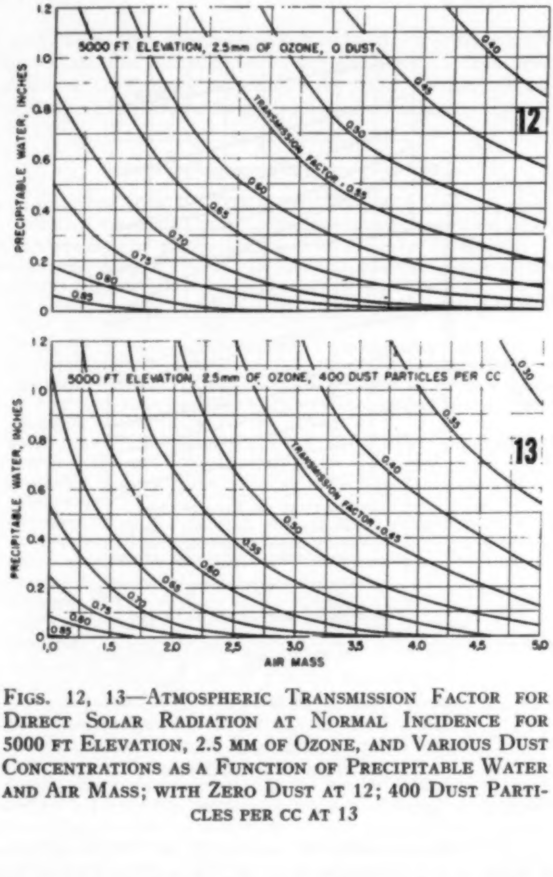


FIG. 11—ATMOSPHERIC TRANSMISSION FACTOR FOR DIRECT SOLAR RADIATION AT NORMAL INCIDENCE FOR SEA LEVEL AND 2.5 MM OF OZONE AS A FUNCTION OF DUST PARTICLES, PRECIPITABLE WATER, AND AIR MASS

A more complete set of calculations were made for 5000-ft elevation. Calculations by the same method previously described for sea level were carried out for the following conditions: 5000 ft elevation; Ozone depth = 2.5 mm; Depth of precipitable water = 0, 10, 30 mm; Dust particles = 0, 400 per cc; Air mass = 1, 2, 3, 4 and 5. Figs. 12 and 13 show the results of the calculations for atmospheric transmission factor for direct solar radiation at normal incidence as a function of precipitable water and air mass for 0 and 400 dust particles per cc, respectively. Detailed comparisons between the sea level calculations and those for 5000-ft

elevation showed that, for other conditions the same, the transmission factors generally agreed within about 3 percent. For example, at air mass 5 and 400 dust particles per cc, the sea level transmission factor differed from that at 5000 ft by 1.4 percent for zero precipitable water, 2.2 percent at 10 mm precipitable water,



FIGS. 12, 13—ATMOSPHERIC TRANSMISSION FACTOR FOR DIRECT SOLAR RADIATION AT NORMAL INCIDENCE FOR 5000 FT ELEVATION, 2.5 MM OF OZONE, AND VARIOUS DUST CONCENTRATIONS AS A FUNCTION OF PRECIPITABLE WATER AND AIR MASS; WITH ZERO DUST AT 12; 400 DUST PARTICLES PER CC AT 13

and 2.9 percent at 30 mm precipitable water. Thus, even at elevations up to 5000 ft the sea level transmission factors are reasonably accurate. This finding is in disagreement with conclusions previously published by others who attributed an increase of 15 percent or more to the effect of an elevation of 5000 ft. The primary reasons for a generally substantial increase of incident radiation at locations at high elevation are the lower depths of precipitable water in the atmosphere,

which is characteristic of the Rocky Mountain states, and the relatively dust-free atmospheres of such localities.

For elevations between 1000 and 5000 ft, linear interpolation between Figs. 8 and 12 and between Figs. 9 and 13 should provide a close estimate of the transmission factor.

It is also of interest to estimate the irradiation of a surface by direct solar radiation at various positions in the atmosphere. Since atmospheric water vapor and

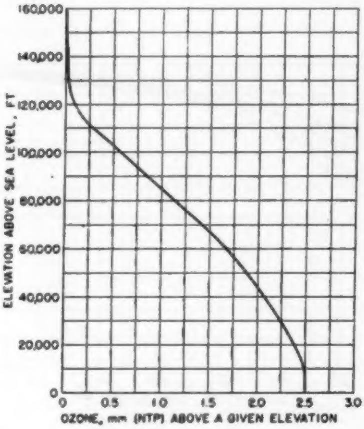


FIG. 14—DISTRIBUTION OF 2.5 MM OF ATMOSPHERIC OZONE (NTP) AS A FUNCTION OF ELEVATION ABOVE SEA LEVEL

dust are primarily concentrated within a few thousand feet of the earth's surface, it is reasonable to assume that at higher location in the atmosphere, depletion of direct solar radiation occurs primarily by dry air scattering and by ozone absorption.

Fig. 14 shows the distribution of 2.5 mm of ozone (NTP) as a function of elevation. Fig. 14 was calculated from data given by Gotz, Meetham, and Dobson.[7]

Calculations were carried out for finding the transmission factor at various positions in the atmosphere assuming no dust and no water vapor and thus based upon depletion by dry air scattering and ozone absorption only. Ozone depths above a given elevation were taken from Fig. 14. The results of the calculations are shown by Fig. 15 which shows the transmission factor as a function of elevation and air mass.

APPLICATION OF TRANSMISSION FACTORS

Before the direct irradiation of a surface at a specific locality and time may be estimated, the sun's position relative to the earth must be known. The sun's

position is established by its declination, altitude, and azimuth angles. The sun's declination which is the angular distance of the sun north or south of the celestial equator is given in *The American Nautical Almanac.*[8] With the declination known, the sun's altitude and azimuth are given in U. S. Hydrographic Office Tables 214.[9] These angles may also be found with less exactness from other sources.[10]

The term *air mass* as used throughout this paper is the optical value and is the ratio of the actual length of path of the sun's rays through the atmosphere to the length of path vertically above the surface in question. At air mass 1, the sun is at the zenith. As used here the air mass is independent of position above sea level.

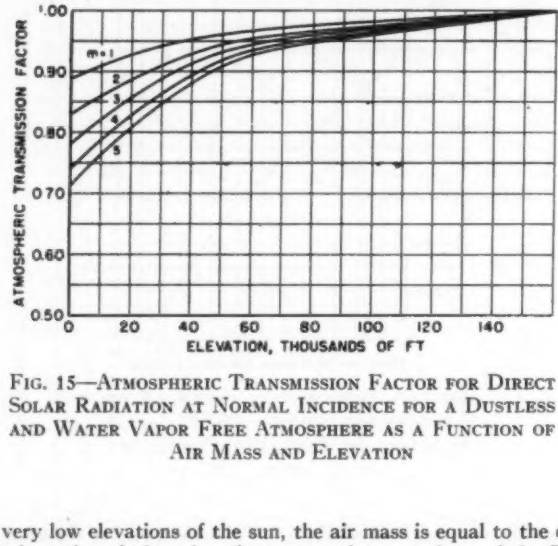


FIG. 15—ATMOSPHERIC TRANSMISSION FACTOR FOR DIRECT SOLAR RADIATION AT NORMAL INCIDENCE FOR A DUSTLESS AND WATER VAPOR FREE ATMOSPHERE AS A FUNCTION OF AIR MASS AND ELEVATION

Except at very low elevations of the sun, the air mass is equal to the cosecant of of the altitude angle. At low elevations, corrections may be made by Bemporad's relation.[11] Fig. 16 shows the relationship between air mass and the sun's altitude angle.

The radiation intensity outside the atmosphere is variable because of the changing distance between the earth and the sun due to the earth's orbit being slightly elliptical, with the sun located off-center of the ellipse. Fig. 17 shows the factor to be multiplied by the solar constant to give radiation intensity at normal incidence at the outer limit of the atmosphere. Fig. 17 is based upon Fowle's data.[12]

Weather Bureau Technical Paper No. 10[13] presents information on mean precipitable water for the United States. Although based on an average of only about 4 years of data per station, it is the most complete source of information available. Figs. 18, 19, 20, and 21 show mean precipitable water in inches between the surface and 8 km altitude for the months of January, April, July, and October, respectively. These figures were reproduced from *Technical Paper No.* 10.[13] The values shown are means for all types of weather conditions. Fritz[14] has shown that during cloudless days the mean precipitable water is approximately 0.85

of that for all days. Thus the values given by Figs. 18-21 should be multiplied by 0.85 for clear-day calculations.

The irradiation of any surface on the earth by direct solar radiation may be calculated if the radiation at normal incidence is known. Fig. 22 defines the angles used in the following equations.

For a surface tilted from the vertical position by an angle, φ the intensity of incident direct solar radiation is given by

$$I_{TV} = I_N (\sin\beta \sin\phi + \cos\beta \cos\alpha \cos\phi) \quad \ldots \quad (3)$$

If the surface is vertical ($\varphi = 0$).

$$I_v = I_N \cos\beta \cos\alpha \quad \ldots \quad (4)$$

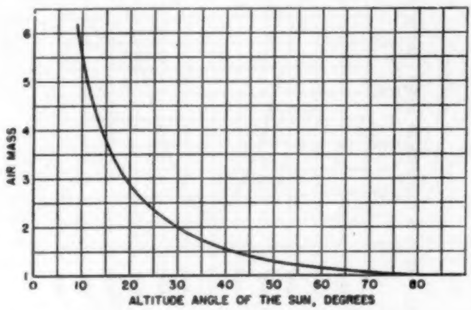


FIG. 16—RELATIONSHIP BETWEEN AIR MASS AND ALTITUDE ANGLE OF THE SUN

If the surface is horizontal ($\varphi = 90$ deg).

$$I_H = I_N \sin\beta \quad \ldots \quad (5)$$

Several examples will now be shown to illustrate use of the information given previously.

Example 1: ***Based upon average conditions, estimate the incidence of direct solar radiation upon a south-facing surface tilted by* 40 *deg from the vertical position at Albuquerque, N. Mex., for* 11:00 *a.m. sun time on a clear July* 15.**

Solution: It will be assumed that Albuquerque is a relatively dust-free location with 200 dust particles per cc. By Fig. 20, $w = 0.95$ in. For clear days, $w = (0.85)(0.95) = 0.81$ in. By Reference 9, $\beta^* = 71° 09'$, $\gamma = 131.8°$; thus $\alpha = 48.2°$ for a south-facing surface. By Fig. 16, air mass = 1.07. Since the elevation above sea level at Albuquerque is 5314 ft, the transmission factor charts for 5000 ft will be used. By Fig. 12 ($d = 0$, $w = 0.81$, $m = 1.07$), $TF = 0.70$. By Fig. 13 ($d = 400$, $w = 0.81$, $m = 1.07$), $TF = 0.66$; thus, $TF = 0.68$ for $d = 200$. By Fig. 17, $I_o/I_{sc} = 0.968$; and

*For meaning of β, γ and α see Fig. 22.

thus, I_N = (0.68) (0.968) (442.4) = 291 Btu per (hr) (sq ft). By Equation 3, I_{TV} = 291 [(0.946)(0.643) + (0.323)(0.667)(0.766)] = 225 Btu per (hr) (sq ft).

Example 2: *Based upon average conditions, estimate the incidence of direct solar radiation upon an east-facing vertical wall at Lincoln, Nebr., for* 11:00 *a.m. sun time on a clear July* 15.

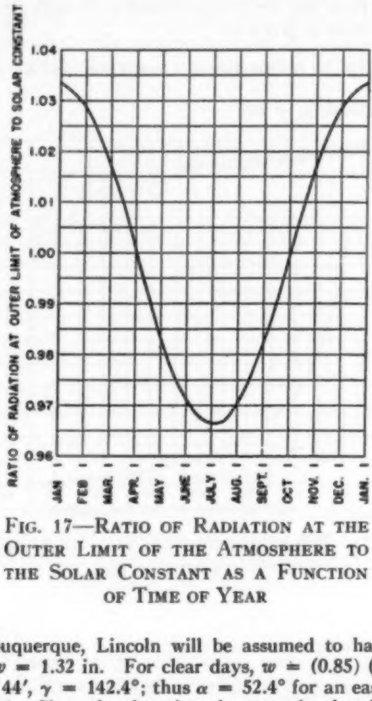


FIG. 17—RATIO OF RADIATION AT THE OUTER LIMIT OF THE ATMOSPHERE TO THE SOLAR CONSTANT AS A FUNCTION OF TIME OF YEAR

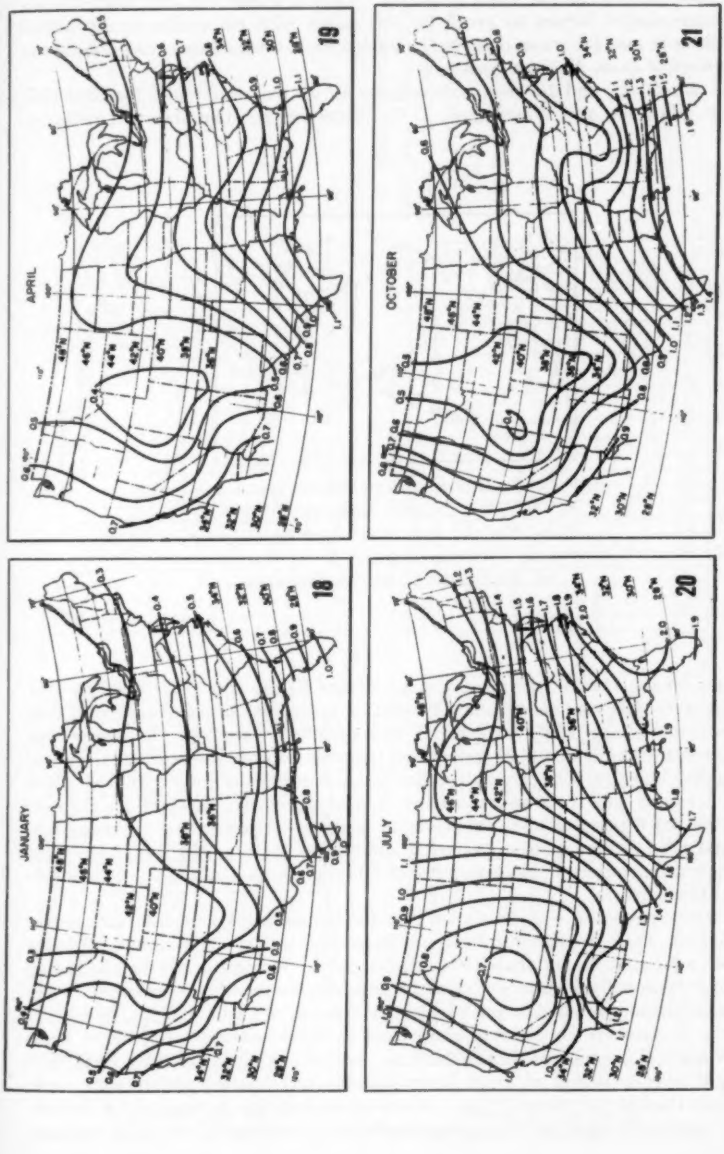
Solution: As for Albuquerque, Lincoln will be assumed to have 200 dust particles per cc. By Fig. 20, w = 1.32 in. For clear days, $w \doteq$ (0.85) (1.32) = 1.12 in. By Reference 9, β = 66° 44′, γ = 142.4°; thus α = 52.4° for an east-facing surface. By Fig. 16, air mass = 1.1. Since the elevation above sea level at Lincoln is 1250 ft, the transmission charts at sea level should be sufficiently accurate. By Fig. 8 (d = 0, w = 1.12, m = 1.1), TF = 0.67. By Fig. 9 (d = 400, w = 1.12, m = 1.1), TF = 0.62; thus TF = 0.645 for d = 200. I_N = (0.645) (0.968) (442.4) = 276 Btu per (hr) (sq ft). By Equation 4, I_v = (276) (0.395) (0.610) = 67 Btu per (hr) (sq ft).

Example 3: *Estimate the incidence of direct solar radiation upon a horizontal surface of an airplane flying at* 50,000 *ft altitude over Lincoln, Nebr., at* 11:00 *a.m. on July* 15.

Solution: By Example 2: β = 66° 44′, air mass = 1.1. By Fig. 15, TF = 0.96. I_N = (0.96) (0.968) (442.4) = 411 Btu per (hr) (sq ft). By Equation 5, I_H = (411) (0.919) = 377 Btu per (hr) (sq ft).

COMPARISON OF CALCULATED AND OBSERVED DIRECT SOLAR RADIATION

The U.S. Weather Bureau has made regular observations of direct solar radiation at normal incidence during clear days at several localities for many years. Hand[15] has described the methods used at the various stations.



FIGS. 18, 19, 20, 21—MEAN PRECIPITABLE WATER IN THE UNITED STATES FOR: JANUARY AT 18; APRIL AT 19; JULY AT 20; OCTOBER AT 21; (BASED ON MAPS SHOWN IN WEATHER BUREAU TECHNICAL PAPER NO. 10)

Comparisons were made between calculated radiation using the mean precipitable water as 0.85 of the value given by *Technical Paper No. 10*[13] with the sea level transmission factors as given by this paper with correction to the actual solar distance and the mean observed radiation for several stations as published in *Climatological Data, National Summary*.

Figs. 23, 24, 25, and 26, show comparisons for air masses 2 and 4 for Blue Hill, Lincoln, Madison, and Washington, D.C., respectively. The observed radiation

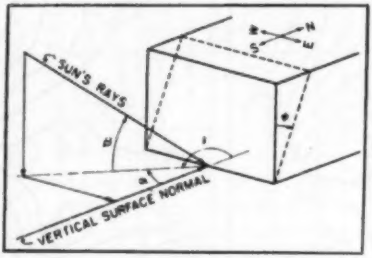
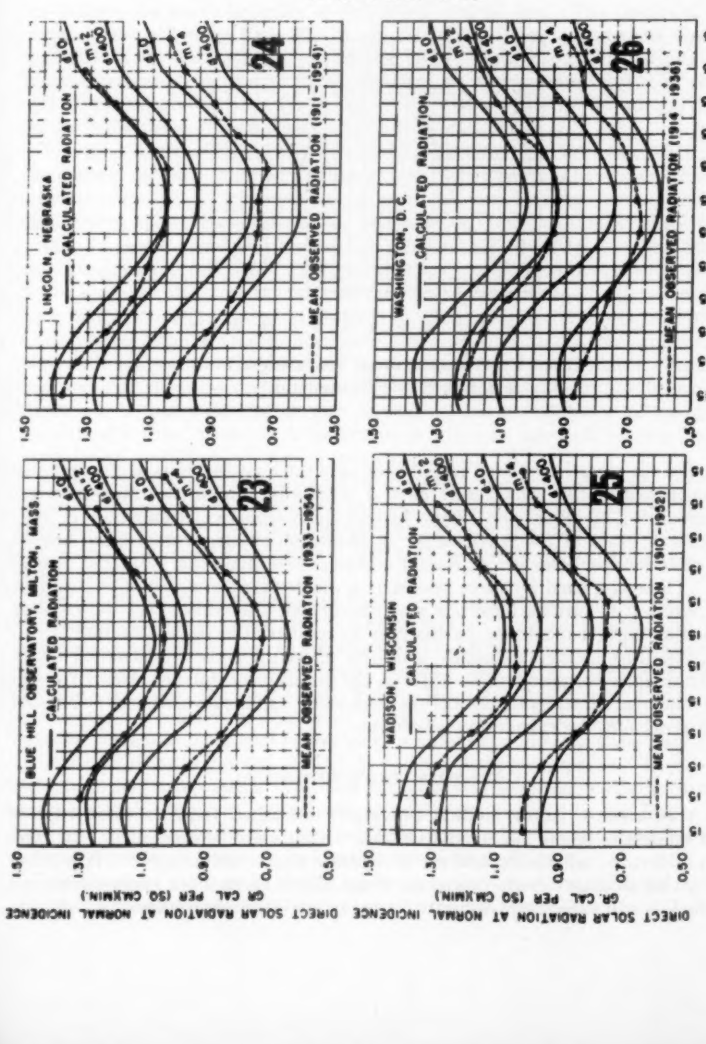


FIG. 22—DEFINITION OF SOLAR ANGLES. α = WALL SOLAR AZIMUTH ANGLE; β = SOLAR ALTITUDE ANGLE; γ = SOLAR AZIMUTH ANGLE; Φ = ANGLE OF TILT FROM VERTICAL POSITION

is shown for the 15th of each month only. These figures show that for Blue Hill, Lincoln, and Madison, the calculated radiation compares quite favorably if a dust concentration of about 200 particles per cc is assumed. However, in interpreting this concentration, it should be recognized that this number is essentially a dust or a turbidity index rather than an absolute concentration since the actual numbers of particles per unit volume are usually defined in relative rather than absolute terms and are dependent upon the efficiency of the collecting device, the magnification and type of illumination employed and similar factors. In this case a dust particle concentration or index number of 200 represents a relatively uncontaminated atmosphere.

On the other hand, Fig. 26 shows that for the dust scale used in this paper, Washington, D.C., appeared to have on the average a dust concentration or index number somewhat higher than 400 particles per cc during the winter and spring seasons. Seasonally high dust concentrations are frequently experienced in the winter and spring periods because of the large volumes of particulate matter ejected into the atmosphere by heating plants during these seasons. Therefore, these results are not unexpected, particularly in view of the fact that these data were averaged over a period of years when coal-burning heating plants were more prevalent than at the present time. A similar analysis was attempted for Boston, Mass. Available observed data were insufficient to permit an adequate compari-



FIGS. 23, 24, 25, 26—COMPARISON OF CALCULATED AND MEAN OBSERVED DIRECT SOLAR RADIATION AT: BLUE HILL OBSERVATORY AT 23; LINCOLN, NEBR., AT 24; MADISON, WIS., AT 25; WASHINGTON, D.C., AT 26

son, but for the data available correlation with calculated radiation occurred with a dust index or concentration of 400 to 600 particles per cc.

The authors believe that the calculation method given in this paper is a reliable procedure for estimating incidence of direct solar radiation. If the atmospheric conditions of dust and water vapor are known, the direct irradiation of a surface may be closely estimated. Precipitable water depth and dust content may vary widely from day to day. As indicated earlier, the estimation of the dust content is the most difficult part of the calculation procedure. An inspection of Equation 2 shows that dust is assumed as the only atmospheric contaminant, but actually since atmospheric dust is less well defined and evaluated than the other factors

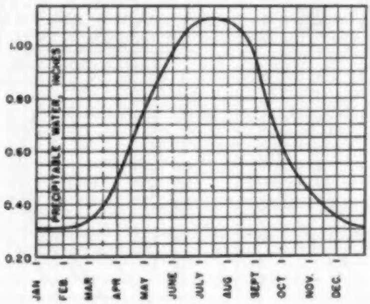


FIG. 27—PRECIPITABLE WATER VARIATION FOR BASIC ATMOSPHERE

involved in the equation, it may include not only the dust turbidity factor but any other neglected factors which may have a small effect upon the transmissivity. Predominantly, however, the factor involves atmospheric cleanliness. A concentration or index of 200 particles per cc should represent a rather clean atmosphere while 800 particles per cc should represent essentially an industrial atmosphere. There is a need for more research on quantitative evaluation of this dust index number in various localities and concurrent observations of direct solar radiation at normal incidence, particularly in industrial localities. While, fortunately, this dust factor has usually a smaller effect upon the transmission factor than the quantity of precipitable water in the atmosphere, further research would permit a more accurate evaluation of the dust index number as used here and would allow extension of the calculation procedure.

APPROXIMATE PROCEDURE

The transmission factor method previously described requires knowledge of several quantities before the irradiation of a surface may be estimated. In addition, considerable calculation and use of separate charts are required. It is possible to devise an approximate procedure which allows much more rapid calculation and which is still sufficiently accurate for many engineering problems.

It has been shown previously that the sea level transmission factor charts are generally applicable to most parts of the United States and that a dust index of 200 particles per cc appears typical for a non-industrial atmosphere. Thus, precipitable water depth is the important variable with seasons of the year and between different parts of the country.

In order to simplify the procedure, it was decided to define a basic atmosphere at a sea level location consisting of 2.5 mm of ozone, 200 dust particles per cc and a varying precipitable water depth as given by Fig. 27. This figure was constructed so as to make the calculated direct solar radiation values closely correspond with the mean observed values at Blue Hill, Lincoln, and Madison throughout the year.

Calculations were then made using appropriate transmission factors for this basic atmosphere for north latitudes of 30, 36, 42, and 48 deg. The results of the calculations are shown in Figs. 28, 29, 30, and 31. These figures show direct solar radiation at normal incidence in Btu per (hour) (square foot) for various daylight hours throughout the year. It is interesting to observe that except at relatively early morning and late afternoon hours, maximum radiation at normal incidence occurs in March and April throughout the country.

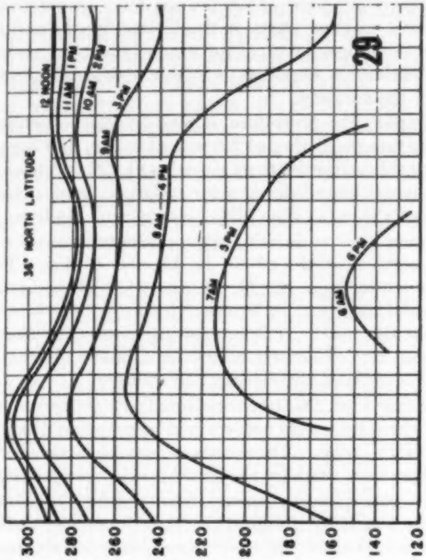
It is proposed that Figs. 28-31 be considered as standard curves and that the basic atmosphere for which they were constructed be considered as having an atmospheric clearness number of unity. These curves may then be used for any locality by multiplying values from them by the atmospheric clearness number of the location.

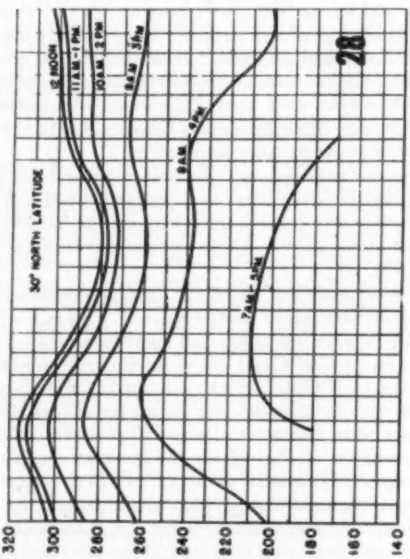
Fig. 32 shows atmospheric clearness numbers for Madison, Lincoln, Blue Hill, and Albuquerque found by dividing the mean observed radiation at normal incidence by that calculated for the basic atmosphere for air mass $1(\beta = 90$ deg), air mass $2(\beta = 30$ deg), and air mass $3(\beta = 19$ deg). Fig. 32 shows that Madison, Lincoln, and Blue Hill have clearness numbers close to unity throughout the year with little scatter due to optical air mass. However, Fig. 32 shows that at Albuquerque, the clearness number varies considerably throughout the year and that there is a considerable scatter with optical air mass. Variation of the clearness number with time of year at Albuquerque is primarily due to the non-proportional variation of precipitable water as compared to Fig. 27.

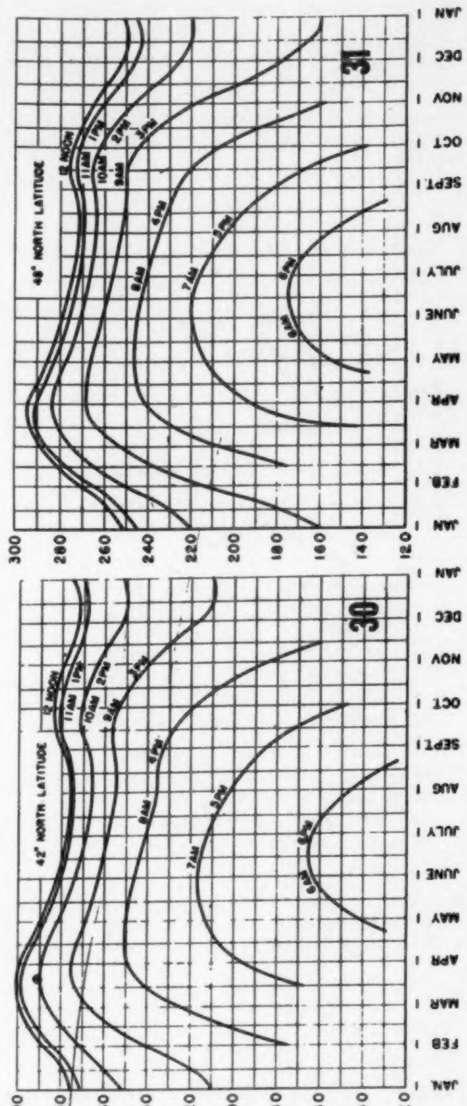
Several calculations for atmospheric clearness number were made for various localities where recorded radiation was not available by dividing the calculated radiation based upon clear-day precipitable water for the locality by the calculated radiation for the basic atmosphere. These calculations together with Fig. 32 were used in constructing Fig. 33.

Fig. 33 shows a broad estimate based on average conditions of the probable atmospheric clearness number in the United States. This map applies only to relatively clean atmospheres as in suburban and rural localities. In the eastern half of the country, except on the Gulf Coast, the same clearness number applies throughout the year. For the Gulf Coast, Rocky Mountain, and Pacific Coast regions, the atmospheric clearness number is lower in winter. This situation results because of the nonproportional character of seasonal precipitable water variation for these localities as compared to Fig. 27 for the basic atmosphere. In the southeastern states, atmospheric clearness numbers less than unity result because of relatively high concentrations of atmospheric water vapor. In the Rocky Mountain states, atmospheric clearness numbers greater than unity occur because of relatively low water vapor concentrations and higher altitudes.

Two examples will now be shown to illustrate the use of the radiation curves for the basic atmosphere and Fig. 33.


29
12 NOON
11 AM
1 PM
10 AM
2 PM
9 AM
3 PM
8 AM — 4 PM
7 AM
6 AM
6 PM
300
280
260
240
220
200
180
160
140
120


28
30° NORTH LATITUDE
12 NOON
11 AM - 1 PM
10 AM 2 PM
9 AM 3 PM
8 AM 4 PM
7 AM - 5 PM
320
300
280
260
240
220
200
180
160
140
DIRECT SOLAR RADIATION AT NORMAL INCIDENCE
BTU PER (HR) (SQ FT)



FIGS. 28, 29, 30, 31—DIRECT SOLAR RADIATION AT NORMAL INCIDENCE WITH THE ATMOSPHERIC CLEARNESS NUMBER OF UNITY FOR VARIOUS NORTH LATITUDES: 30 DEG AT 28; 36 DEG AT 29; 42 DEG AT 30; 48 DEG AT 31

Example 4: *Rework Example 2 by approximate procedure.*

Solution: The latitude at Lincoln is 40° 49′. By Fig. 30, I_N = 276 Btu per (hr) (sq ft) for basic atmosphere. By Fig. 33, *C.N.* = 1.0. By Equation 4. I_v = (276) (0.395) (0.610) = 67 Btu per (hr) (sq ft).

Example 5: *Based upon average conditions compare the incidence of direct solar radiation upon a flat roof of a building in suburban Minneapolis with one in Dallas at 12:00 noon sun time on a clear July 1st.*

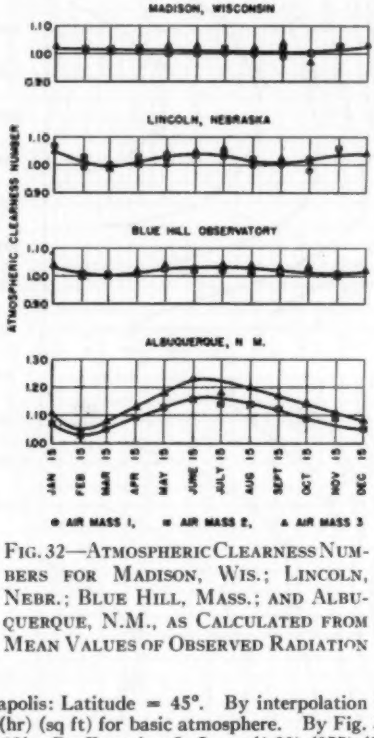


FIG. 32—ATMOSPHERIC CLEARNESS NUMBERS FOR MADISON, WIS.; LINCOLN, NEBR.; BLUE HILL, MASS.; AND ALBUQUERQUE, N.M., AS CALCULATED FROM MEAN VALUES OF OBSERVED RADIATION

Solution: For Minneapolis: Latitude = 45°. By interpolation between Figs. 30 and 31, I_N = 277 Btu per (hr) (sq ft) for basic atmosphere. By Fig. 33, *C.N.* = 1.02. By Reference 9, β = 68° 10′. By Equation 5, I_H = (1.02) (277) (0.928) = 262 Btu per (hr) (sq ft).

For Dallas: Latitude = 33°, I_N = 280 Btu per (hr) (sq ft) for basic atmosphere. By Fig. 33, *C.N.* = 0.95. By Reference 9, β = 80° 10′. By Equation 5, I_H = (0.95) (280) (0.985) = 262 Btu per (hr) (sq ft). Thus the direct radiation is the same for each locality.

The approximate nature of the clearness number method of calculation and its limitation to broad estimates of direct solar radiation during clear days for relatively clean atmospheres should be recognized. For industrial localities, the atmospheric clearness number is less than that given by Fig. 33. As stated earlier, there is a need

for more experimental observations of direct solar radiation, particularly in industrial localities. Any such observations may be immediately expressed in terms of atmospheric clearness numbers for the locality. Thus the clearness number concept appears to be a convenient way of correlating experimental data. The authors believe that this approach provides a simple and reasonably accurate method of estimating for engineering purposes the incidence of direct solar radiation during clear days.

Acknowledgments

The authors wish to acknowledge the assistance of Harrison Benjamin, graduate research assistant, Department of Mechanical Engineering, University of Minne-

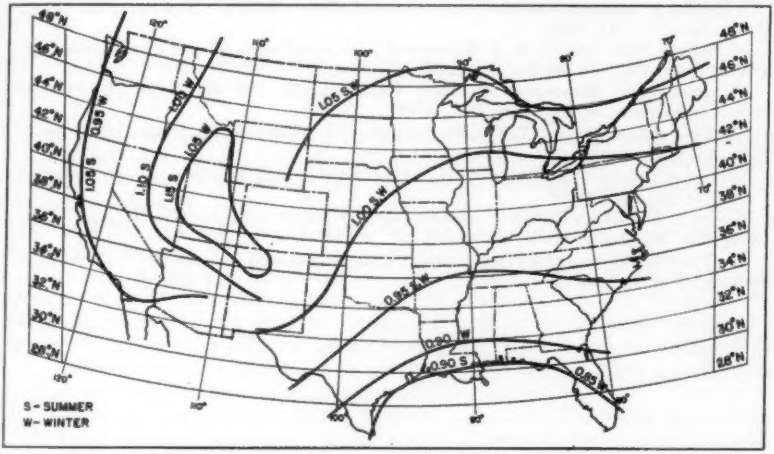


Fig. 33—Estimated Atmospheric Clearness Numbers in the U.S. for Non-industrial Localities

sota, during the research which resulted in this paper. Mr. Benjamin was responsible for most of the calculations and in addition contributed many original ideas which were incorporated into the paper.

The authors also wish to acknowledge the assistance of Robert H. Hanson, research assistant, Department of Mechanical Engineering, University of Minnesota, who made many of the calculations.

In addition, the authors wish to express their appreciation to the National Science Foundation which provided the financial support for the research.

References

1. Proceedings of the 1957 Solar Furnace Symposium (*The Journal of Solar Energy Science and Engineering*, Vol. 1, Nos. 2, 3, April-July 1957).

2. Performance of a Solar Heated Office Building, by F. H. Bridgers, D. D. Paxton and R. W. Haines (ASHAE TRANSACTIONS, Vol. 64, 1958, p. 83).

3. The Solar Constant, by F. S. Johnson (*Journal of Meteorology*, Vol. 11, December 1954, pp. 431-439).

4. Solar Energy on Clear and Cloudy Days, by Sigmund Fritz (*The Scientific Monthly*, Vol. 84, February 1957, pp. 55-56).

5. Proposed Standard Radiation Curves for Engineering Use, by Parry Moon (*Journal of the Franklin Institute*, Vol. 230, November 1940, pp. 583-617).

6. The Transparency of Aqueous Vapor, by F. E. Fowle (*Astrophysical Journal*, Vol. 42, 1915, p. 397).

7. The Vertical Distribution of Ozone in the Atmosphere, by F. W. P. Gotz, A. R. Meetham and G. M. B. Dobson (*Proceedings of the Royal Society of London*, Series A, Vol. 145, 1934, pp. 416-446).

8. *The American Nautical Almanac* (U.S. Naval Observatory, Washington, D.C., Annual).

9. *Tables of Computed Altitude and Azimuth*, Vol. 1-9 (U.S. Hydrographic Office, Publication No. 214, 1940).

10. Where Is the Sun, by W. J. Wilson and J. M. Van Swaay (*Heating and Ventilating*, Vol. 39, May 1942, pp. 41-46; June 1942, pp. 59-61).

11. *Smithsonian Meteorological Tables*, Fifth Edition, Vol. 86, Table 100 (Smithsonian Institution, Washington, D.C., 1931).

12. *Smithsonian Physical Tables*, Sixth Edition (Smithsonian Institution, Washington, D.C., 1914, Table 181).

13. Mean Precipitable Water in the United States (U.S. Weather Bureau *Technical Paper No.* 10, April 1949).

14. Solar Radiation During Cloudless Days, by Sigmund Fritz (*Heating and Ventilating*, Vol. 46, January 1949, pp. 69-74).

15. Pyrheliometers and Pyrheliometric Measurements, by I. F. Hand (U.S. Weather Bureau, Washington, D.C., 1946).

1623

No. 1623

A HIGH-FLUX LOW-TEMPERATURE SOLAR COLLECTOR

By R. G. Nevins*, Manhattan, Kans., and P. E. McNall, Jr.**, Hopkins, Minn.

DURING the past few years numerous technical articles have been published dealing with the utilization of solar energy for house heating. In general, these articles have dealt with the problems involved in the design and construction of flat-plate collectors. These collectors are not complicated and are fairly inexpensive to build. They are not equipped with any type of focusing or concentrating apparatus (Fig. 1).

The classical paper of Hottel and Woertz[1] published in 1942 was an analysis of the design and performance of flat-plate collectors. In 1955, H. Tabor at the First World Symposium of Solar Energy and Scientific Conference presented 2 papers[2,3] dealing with collector design with special reference to the principle of selective radiation (the possibility of producing a surface having a low effective emissivity which is also a good absorber of solar energy).

The principle involved in collector design is to construct a collector plate which will absorb the maximum amount of solar energy possible into the working fluid with a minimum loss to the environment. Hottel and Woertz made use of the greenhouse effect of plate glass and the insulating effect of 1 or 2 air spaces to reduce the loss to the environment. Tabor suggests the use of a collector plate whose surface has a low emissivity in the 3-25 μ range but which is a good absorber of solar energy. Materials with these properties are not readily available and much research is needed to perfect materials of this type. A third method of reducing this loss is to reduce the heat transfer area exposed to the environment but maintain the same amount of incident radiation by the use of a suitable concentrating reflector (Fig. 1); the collector temperature being maintained at a relatively low level.

A study of this third method of improving solar collection efficiencies was undertaken during the summer of 1956, and experimental data were obtained using a

---

*Professor and Head, Department of Mechanical Engineering, Kansas State College. Associate Member of ASHAE.

**Chief Engineer, Applied Research Department, Temperature Controls Group, Minneapolis-Honeywell Regulator Co. Associate Member of ASHAE.

[1]Exponent numerals refer to References.

Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, Pa., January 1958.

surplus army searchlight reflector. The results of the study and the data obtained are presented herein.

ANALYSIS

The energy balance for a solar collector under equilibrium conditions is:

$$\alpha\beta PQ_i = q_{eT} + q_{sT} \quad \dots \quad (1)$$

where

α = absorptivity of the collector plate.
β** = transmissivity or reflectivity of the focusing system or cover plates.
P = concentrating power—ratio of the area of intercepted solar radiation to the area of the collector surface.
Q_i = incident solar radiation, Btu per (hour) (square foot).
q_{eT} = energy loss to the environment by radiation, convection and conduction per unit area of the collector at temperature T, Btu per (hour) (square foot).
q_{sT} = rate of useful energy extracted from the collector at temperature T, per unit area of the collector, Btu per (hour) (square foot).

For a flat-plate collector $P = 1$. If the collector efficiency is defined as the ratio of the useful energy extracted from the collector, q_{sT}, to the energy incident on the collector, PQ_i,

$$\eta_T = \alpha\beta[1 - (q_{eT}/\alpha\beta PQ_i)] \quad \dots \quad (2)$$

where

η_T = collection efficiency at temperature, T.

The maximum efficiency of a collector is determined by its absorptivity, α, and the transmissivity, β, of the focusing system. From Equation 2 it is evident that the maximum efficiency can be approached by reducing q_{eT} or increasing P, or both.

The heat loss to the environment from a collector plate, assuming an average temperature for the collector, is:

$$q_{eT} = h(T_c - T_a) + \epsilon_c\sigma[T_c^4 - T_a^4] + U_b(T_c - T_a) \quad \dots \quad (3)$$

where

q_{eT} = energy loss to the environment per unit area of collector, Btu per (hour) (square foot).
h = convective film coefficient, Btu per (hour) (square foot) (Fahrenheit degree) [mainly a function of collector size and wind velocity].
T_c = collector temperature, Rankine.
T_a = environmental temperature, Rankine (Usually dry-bulb temperature of the air).
ϵ_c = emissivity of the collector surface.
σ = Stefan-Boltzmann constant.
U_b = back loss coefficient, Btu per (hr) (square foot of collector area) (Fahrenheit degree).

For a collector equipped with a glass cover plate q_{eT} is:

$$q_{eT} = h_s(T_c - T_g) + F\sigma[T_c^4 - T_g^4] + U_b(T_c - T_a) \quad \dots \quad (4)$$

and

**β is the mirror reflectivity for a mirror system or the transmissivity of the glass cover plates for a flat-plate collector or the product of the two if both are involved.

$$q_{oT} = h(T_g - T_a) + \epsilon_g\sigma[T_g^4 - T_a^4] + U_b(T_c - T_a) \quad . \; . \; . \; . \; . \quad (5)$$

where

h_s = convective heat transfer coefficient for air space created by collector plate and glass cover, Btu per (hour) (square foot) (Fahrenheit degree).

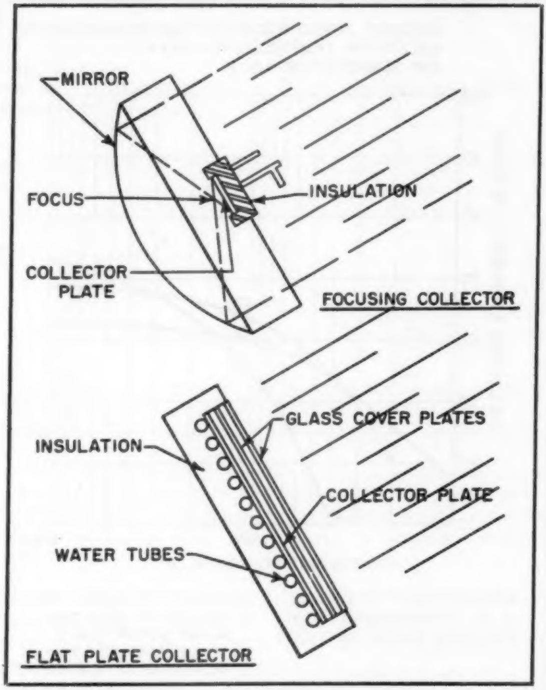


Fig. 1—Focusing Collector with Collector Mounted Behind the Mirror Focus. Flat-plate Collector with 2 Glass Cover Plates

T_g = glass temperature, Rankine.
F = radiant interchange factor between two parallel surfaces, dimensionless $= 1/(1/\epsilon_c + 1/\epsilon_g - 1)$.
ϵ_g = emissivity of glass.
h = convective heat transfer coefficient from outermost plate to the environment, Btu per (hour) (square foot) (Fahrenheit degree).

For two glass cover plates an additional equation can be written for the heat loss. The first 2 terms in Equation 4 represent the heat transferred between the collector

plate and the first glass cover and in Equation 5 the first 2 terms represent the heat transferred between the last glass cover and the surrounding air. The third term in both cases is the back loss. For 2 glass covers an equation similar to Equation

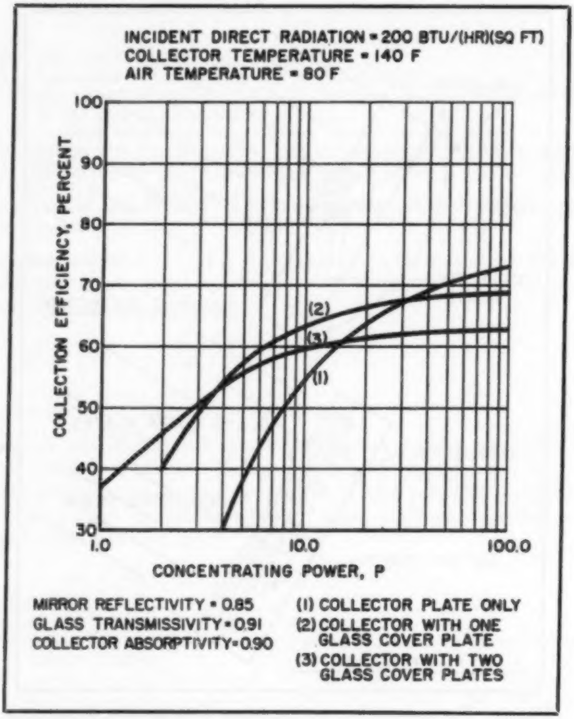


FIG. 2—CALCULATED COLLECTION EFFICIENCIES OF A SOLAR COLLECTION SYSTEM AS A FUNCTION OF THE CONCENTRATING POWER, P, CONSTANT RECEIVING AREA OF 19.6 SQ FT

4 could be written for the heat transfer between the first glass cover plate and the second or last plate. To determine the heat loss in the latter two cases, the equations are solved simultaneously to eliminate the unknown glass temperatures.

Using the equations just given, the collection efficiency for various values of P was calculated for a solar collector system having a constant receiving area of 19.6

sq ft. (Receiving area times the incident solar radiation gives the total solar energy entering the collecting apparatus). These results based on the following assumptions are shown in Fig. 2.

Collector temperature, 140 F.
Air temperature, 80 F.
Incident solar radiation, 200 Btu per sq ft.
Reflectivity of the mirror, 0.85.
Absorptivity of the collector, 0.90.
Transmissivity of the glass cover plate, 0.91.
Emissivity of the glass cover plate, 0.95.
Glass spacing, ⅛ in.
Overall coefficient of heat transmission U_b for the back loss, 0.194 Btu per (hr) (sq ft of collector area) (F deg).

TABLE 1—MAXIMUM THEORETICAL COLLECTION EFFICIENCIES[a]

COVER PLATES	FLAT PLATE COLLECTORS NO FOCUSING SYSTEM	FOCUSING COLLECTOR
None	0.900	0.765
One	0.819	0.696
Two	0.745	0.633

[a]Heat Loss = 0; collector absorptivity = 0.90; glass transmissivity = 0.91; reflection coefficient = 0.85.

Film coefficient of heat transfer h for the collector was calculated from the equation, given by Jakob, $N_{Nu} = 0.592\ R_e^{1/2}$ where $N_{Nu} = hL/K$, $R_e = VL\ \rho/\mu$. The radius of the collector is used as the characteristic length. Velocity of the wind was 10 mph.

Film coefficient of heat transfer h_s between the cover plates was calculated from the equation given by McAdams, $h_s = K/L$ where K is the conductivity of the air in the space and L is the width of the space. The equation is valid for natural convection in sealed spaces for $N_{Gr}N_{Pr} < 10^3$ where N_{Gr} is the Grashof number and N_{Pr} is the Prandtl number.

The calculations show that a collector with 2 glass cover plates (low heat loss) has higher collection efficiencies up to $P = 3$, a collector with 1 glass cover plate is more efficient over the range $P = 3$ to $P = 30$ and a collector with no cover plate is more efficient for values of P greater than 30.

The maximum theoretical efficiencies obtainable are given in Table 1 and are presented to show the maximum values which would be attainable assuming the heat loss to the environment, q_{eT}, equal to zero. The values given in the second column are the asymptotes for the curves in Fig. 2.

The table shows the flat-plate collector to be theoretically superior if the heat loss is zero, due to the assumed 85 percent reflectivity of a suitable concentrating mirror. Practical considerations, as taken into account in Fig. 2, indicate that the focusing collector makes possible a much greater reduction in the heat loss than is possible with the flat-plate collector.

EXPERIMENTAL COLLECTOR

Using a surplus army searchlight reflector as the focusing system, a solar collector was designed and built so that the value of P was 97. The receiving area of the

system, equal to the area of the searchlight minus the area of the collector, was approximately 19.6 sq ft.

The collector (see Fig. 3) consisted of a collector plate of 6-in. diam which could be cooled by circulating water from the center to the outer edge through a spiral

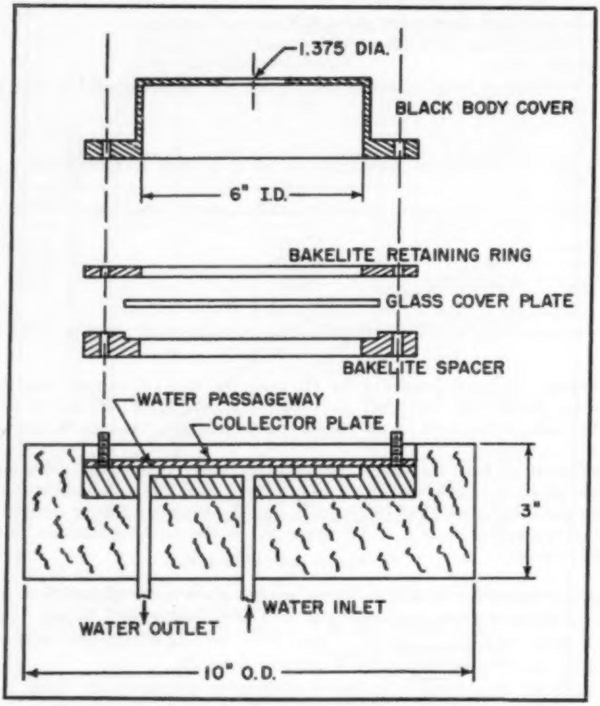


FIG. 3—6-IN. DIAM COLLECTOR USED WITHOUT COVER, WITH 1 GLASS COVER, OR WITH BLACK BODY COVER

passageway. The front surface of the plate was painted black and later blackened further with acetylene soot. The edge and back of the collector were insulated with glass fiber insulation. The entire assembly was 10 in. in diameter and 3 in. deep.

The collector assembly was mounted in the 5-ft diam reflector and positioned so that the image completely filled the 6-in. diam collector plate (shown schemati-

cally in Fig. 1). The plate was located either in front of or behind the focus of the mirror. Data were obtained with the plate in both positions, but as was expected no effects due to the position could be detected.

In using the collector, the searchlight mirror was directed at the sun and maintained in this position manually. Cooling water was pumped through the collector so that a temperature rise of approximately 5 F deg was maintained. The water

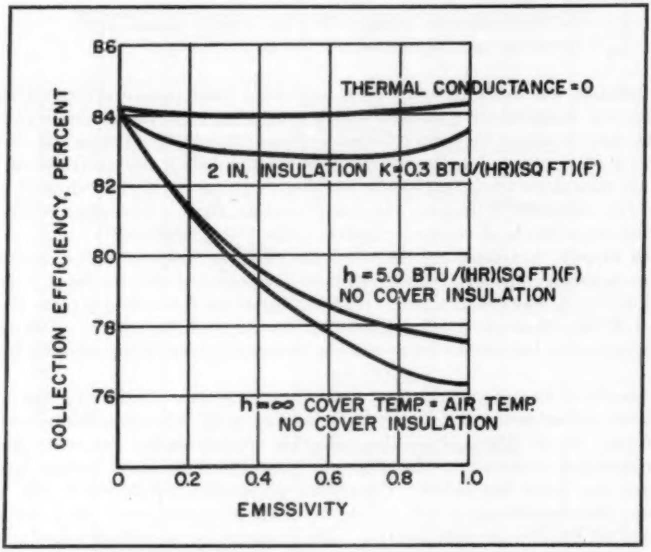


FIG. 4—EFFECT OF COVER EMISSIVITY AND COVER INSULATION ON COLLECTION EFFICIENCY USING A BLACK BODY COVER. 6-IN. DIAM COLLECTOR PLATE IN A 5-FT DIAM FOCUSING MIRROR (P = 97); INCIDENT SOLAR RADIATION = 300 BTU PER (HR) (SQ FT) (F DEG); COVER TEMPERATURE, 140 F; COVER EMISSIVITY, 0.9; AIR TEMPERATURE, 80 F

temperature leaving the collector and the temperature rise were determined by means of thermocouples located at the inlet and outlet of the collector. The water flow rate was approximately 8 lb per min. A pyrheliometer, mounted on the searchlight shell, was used to determine the amount of total radiation received at the mirror. The total radiation and the sky radiation were determined directly and the direct radiation was determined from the difference of the two values. The amount of sky radiation was determined by shading the thermopile in the pyrheliometer from the direct rays of the sun and measuring the thermopile output.

The efficiency of the collector was calculated from instantaneous readings of flow rate, temperature rise, and the amount of direct radiation:

$$\eta = mC\,\Delta\,T/Q_i A_R \quad \ldots \ldots \ldots \ldots \quad (6)$$

where

m = flow rate, pounds per hour.
C = specific heat, Btu per (pound) (Fahrenheit degree).
ΔT = temperature increase, Fahrenheit.
Q_i = incident radiation (direct) Btu per (hour) (square foot).
A_R = receiving area, square feet = 19.6 sq ft.

No attempt was made to maintain a fixed inlet water temperature. Therefore, the collector temperature gradually increased during a particular day's run.

Data were obtained with the collector in front of and behind the focus, with and without a glass cover plate over a range of mean collector temperatures of 100 to 170 F on clear days (data taken only when sun was not covered by clouds). Some data were obtained on cloudy and partly cloudy days. The glass cover plates were cut from sheets of common window glass, ⅛-in. thick.

In an attempt to reduce the reflection and emission loss from the collector plate without reducing the radiation incident on the collector plate, a *black body cover*, shown in Fig. 3, was constructed. When mounted on the collector plate the cover had a 1.375-in. diameter opening located at the focus of the mirror. The collector plate was located behind the focus with the image of the sun filling the 6-in. diameter plate.

A theoretical analysis of the *black body cover* was made considering the multiple reflections and emissions inside the cover assuming all reflections and emissions to be diffuse. From this analysis the collection efficiencies for the cover described were calculated to determine the effect of emissivity of the cover surface facing the collector and cover insulation. Calculated efficiencies are shown in Fig. 4 for a collector plate emissivity of 0.9, a collector plate temperature of 140 F and a solar intensity of 300 Btu per (hr) (sq ft). The geometry of the collector and collection system is assumed identical to that previously described. It is shown that theoretically the efficiency of the focusing collector can be increased with the use of a *black body cover*. If the cover is insulated, the cover emissivity has little effect on the efficiency but if the cover is not insulated, the cover emissivity should be low for maximum efficiency.

RESULTS

The calculated efficiencies for the collector system on relatively clear days are given in Fig. 5. The amount of incident direct radiation varied from approximately 180 to 215 Btu per (hr) (sq ft). Efficiencies of about 55 percent were obtained with the collector equipped with a glass cover plate. Without a glass cover, efficiencies of about 65-73 percent were obtained and with the *black body cover* efficiencies of about 72 to 75 percent were obtained. The efficiencies compare favorably with theoretical values of 68.8 percent for a collector with one glass cover, 73 percent for a collector without a cover, and 81 percent for a collector with the *black body cover*. The theoretical efficiencies were calculated assuming a collector plate emissivity of 0.9 and a collector plate temperature of 140 F. The difference

between the measured and calculated efficiency for the collector with one glass cover plate is believed due to the use of a 91 percent transmissivity for the glass and the experimental difficulties encountered in using the glass cover. The glass, ⅛-in. common window glass, had a tendency to break a short time after exposure

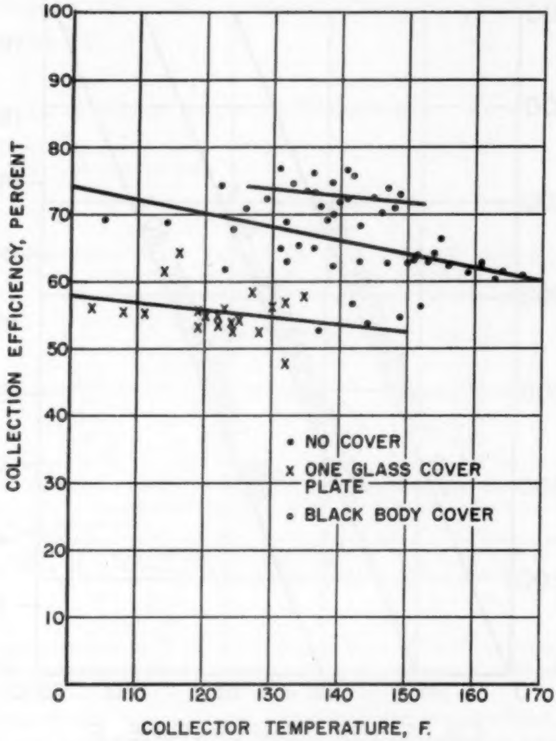


FIG. 5—COLLECTION EFFICIENCIES FOR A 6-IN. DIAM COLLECTOR MOUNTED IN A 5-FT DIAM PARABOLIC MIRROR, CONCENTRATING POWER = 97

to the sun and therefore the data were obtained during relatively short exposure periods.

The effect of a variation in Q_i can be determined from Equation 2. Since Q_i varies from zero to some maximum value during a day, the efficiency of any solar collector will vary likewise. However, for the data presented in Fig. 5, the magnitude of the factors in Equation 2 is such that a variation of ± 10 percent produces

a variation in the efficiency of only ± 0.85 percent. This is true only during that period of the day when Q_i is of about 200 Btu per (hr) (sq ft).

Analysis of the collector to determine the daily collection efficiency or the annual collection efficiency was not made at this time. This analysis requires knowledge

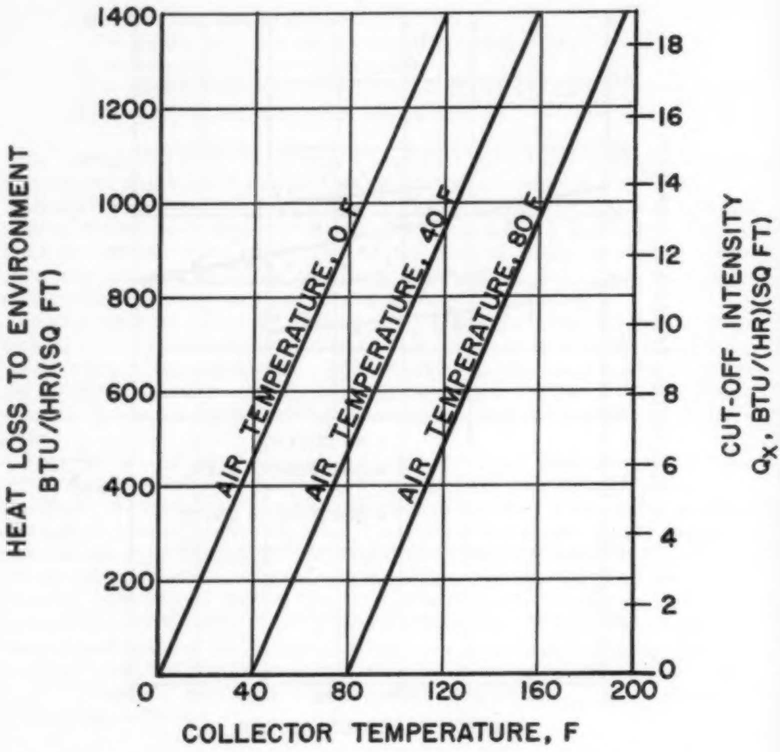


FIG. 6 — CALCULATED VALUES OF CUT-OFF INTENSITY AND HEAT LOSS TO THE ENVIRONMENT FOR A 6-IN. DIAM COLLECTOR (P = 97); h = 10.6 BTU PER (HR) (SQ FT) (F DEG); $\epsilon = \alpha = 0.9$; $\beta = 0.85$; INSULATION THICKNESS 1½ IN.

of the daily radiation curves and daily temperature curves and requires numerous graphical integrations. For flat-plate collectors, Jordan and Threlkeld[4] give daily collector efficiencies for January 15, Lat. 42 deg North for collectors with 1, 2, and 3 glass cover plates for a clear day and for a partly cloudy day. For a clear day and a collector temperature minus air temperature difference of 60 F, they obtain a calculated daily efficiency of 56 percent.

Tabor[2] has calculated the annual collection efficiencies for various collectors, mostly of the flat-plate type. For a flat-plate collector with a mean temperature of 140 F, air temperature 70 F, he calculates an annual efficiency of approximately 55 percent.

An important quantity in the analysis of a solar collector is the so-called *cut-off* or *threshold* value for a particular collector. As can be seen from Equation 2, the collection efficiency will be zero for values of Q_i less than $q_{eT}/\alpha\beta P$ or:

$$Q_x = q_{eT}/\alpha\beta P$$

where

Q_x = cut-off intensity Btu per (hour) (square foot).

i.e. that value of Q_i below which the collector cannot reach temperature T, and no useful collection will be possible.

TABLE 2 — REPRESENTATIVE VALUES OF EXPERIMENTAL COLLECTION EFFICIENCIES ON PARTLY CLOUDY DAYS[a]

COLLECTOR TEMPERATURE F	USEFUL ENERGY BTU/HR	DIRECT RADIATION BTU/HR	COLLECTION EFFICIENCY PERCENT	TOTAL RADIATION BTU/HR	COLLECTION-EFFICIENCY PERCENT
129.3	906	2065	43.9	3171	28.6
136.5	974	2010	49.0	3113	31.3
141.0	1573	2523	62.3	3747	41.9
150.2	1880	3300	57.0	4185	44.9
127.0 av.	494.6[b]	797[b]	62.2	1275[b]	38.8

[a]6-in. diam collector in 5-ft diam parabolic reflector, $P = 97$. No glass cover, collector in back of the focus.
[b]Total energy for 20 min, Btu.
Note: The first 4 items were determined from instantaneous readings of temperature, water rate, and pyrheliometer output. The fifth item was determined from data obtained over a 20-min period during which time the useful energy obtained varied from 470 Btu per hr to 2640 Btu per hr.

The calculated values of Q_x for a focusing collector, $P = 97$, and the calculated values of the heat loss to the environment, q_eT, for air temperatures of 0, 40, and 80 F as a function of collector temperature are given in Fig. 6.

Some representative values for the collection efficiencies calculated from data obtained on partly cloudy days are given in Table 2. In this case both the direct radiation intensity and the total (sky plus direct) radiation intensity were used to calculate the collection efficiencies and the 2 values are compared in Table 2.

It is noted that on clear days during the period of peak radiation intensities the sky radiation is approximately 10 percent of the total radiation, while on cloudy or overcast days the sky radiation is approximately 33 percent of the total radiation. Assuming that only a small portion of the sky radiation is effective in a focusing type collector, the clear day efficiencies are based on the intensity of the direct radiation. If comparisons are to be made with flat-plate collectors, the efficiency should be based on the total radiation especially if the comparison is to be made using data obtained on partly cloudy or cloudy days.

A disadvantage of the focusing collector is that it requires a heliostatic mounting which will keep the plane of the mirror at right angles to the sun's rays at all

times. A flat-plate collector is usually a fixed device, orientated toward the south at the latitude angle from horizontal.

CONCLUSIONS

The collection efficiency of a solar collector has a maximum limit determined by the product of the absorptivity of the collector plate and the transmissivity of any focusing system or glass cover plates (Table 1).

By use of a focusing parabolic mirror these limiting efficiencies can be approached when the value of receiving area to collector area is 50 to 100 or higher, using a low collection temperature of about 150 F. The analysis shows that for values of P greater than 30 no glass cover plates should be used since a collector with no cover plates has the highest limiting efficiency, for glass transmissivities of 0.91 or less.

Collection efficiencies of 65 to 75 percent were obtained using a 6-in. diam collector mounted in a 5-ft diam surplus searchlight (P = 97) on clear days with collector temperatures from 100 to 170 F and air temperatures of approximately 80 F.

Efficiencies for the experimental collector on cloudy days varied considerably but were about 40 to 60 percent based on the intensity of the direct radiation and 30 to 45 percent when based on the intensity of the total radiation.

Although the data were not conclusive, there was some indication that the *black body cover* reduces the heat loss to the environment and thereby tends to increase the collection efficiency. Theoretical analysis of the *black body cover* shows that the maximum theoretical efficiency, for the focusing collector system described in this paper, can be increased from 76.5 percent to 83 to 84 percent by using the cover. Cover emissivity is shown to have little effect on the efficiency if the cover is insulated. For an uninsulated cover, the cover emissivity should be low for maximum efficiency.

ACKNOWLEDGMENT

The authors wish to acknowledge the assistance of Willard Anderson and various other members of the staff at the Honeywell Research Center, Hopkins, Minn.

REFERENCES

1. Performance of Flat Plate Solar Collectors, by H. C. Hottel and B. B. Woertz (*ASME Transactions*, Vol. 64, 1942, pp. 91- 104).

2. Solar Energy Design, by H. Tabor (*Bulletin Research Council of Israel*, Vol. 5C, 1955; presented at the First World Symposium on Solar Energy at Phoenix, Ariz.).

3. Selective Radiation, by H. Tabor (*Bulletin Research Council of Israel*, Vol. 5A, 1955).

4. ASHVE RESEARCH REPORT No. 1502 — Availability and Utilization of Solar Energy, by R. C. Jordan and J. L. Threlkeld (ASHVE TRANSACTIONS, Vol. 60, 1954, pp. 177-238).

DISCUSSION

H. TABOR, Jerusalem,* Israel, (WRITTEN): This paper will be, perhaps, of greater interest to those generally engaged in solar energy exploitation research than to heat-

*The National Physical Laboratory of Israel.

ing and air-conditioning engineers, for it is extremely doubtful if *domestic* solar appliances will ever be heliostatic. There are two reasons for this:

(a) complexity and maintenance problems of the heliostat; and

(b) heliostatic systems cause shadowing over a large area so that if several collectors are required for a given output, a large space must be left between them.

If a heliostatic mounting is permitted, the performance of a flat plate collector is very considerably improved and may be only slightly below that of the focusing collector described at the relatively low temperatures considered. Indeed in cloudy weather, where the flat plate collector can exploit diffuse radiation, its performance can exceed that of the focusing system.

In considering efficiency, it is advisable to consider the efficiency over a whole day (or a whole year) rather than for a short period since, in practice, the wish is to exploit as much of the sunshine as possible. The present paper deals only with instantaneous efficiencies; the daily efficiencies will be slightly less though the difference is far less in the heliostatic — high concentration system than in a flat plate collector. Thus Equation 2 gives the collector efficiency as

$$\eta_T = \alpha\beta[1 - (q_{eT}/\alpha\beta P Q_i)]$$
$$= \alpha\beta\left(1 - \frac{Q_x}{Q_i}\right)$$

where

$$Q_x = q_{eT}/\alpha\beta P = \text{the } \textit{cut off} \text{ intensity.}$$

The quantity $\left(1 - \frac{Q_x}{Q_i}\right)$ has been termed the ***retention*** efficiency (reference 2), since it is the fraction of the absorbed incoming energy retained by the collector plate after deducting the heat losses. When $P = 97$ as in the case considered and q_{eT} is of the order of several hundred Btu per (hr) (sq ft), this yields Q_x values of the order of 10 Btu per (hr) (sq ft) (see Fig. 6) so that the retention efficiency is 0.97 for $Q_i = 300$ Btu per (hr) (sq ft) which would be about the solar intensity at noon on a clear day; this efficiency would drop off only very slowly at other hours, for a heliostatic mounting. The maximum theoretical efficiencies given in Table 1 are based upon a retention efficiency of unity, and simply correspond to the value of the $\alpha\beta$ product.

As a matter of comparative interest, consider a flat plate collector with a collector surface absorptivity α of 0.92, an emissivity e_e of 0.1* and a single glass cover of transmissivity $\beta = 0.91$. The heat loss q_{eT} at 140 F to an environment at 80 F will be about 40 Btu per (hr) (sq ft)†† which leads to a cut-off value of $Q_x = \frac{40}{\alpha\beta} = 48$.

The retention efficiency $\left(1 - \frac{Q_x}{Q_i}\right)$ is thus 0.84 for the noon value of $Q_i = 300$ which is only 12 percent less than the value of 0.97 for the concentrating system.

The treatment of a black body cover is interesting but hides, unwittingly, the fact that the concentration is potentially increased to 1840 because the solar radiation is now concentrated on a circle of 1.375-in. diam instead of 6-in. diam. In other words, the collector plate could have been made 1.375-in. diam instead of 6-in. assuming that this were a practical arrangement. The retention efficiency of course becomes almost unity because of the very large P, and assuming the cover to be reasonably insulated.

---

*Such *selective* surfaces are now being produced in large areas in Israel and arrangements are being made to make them available in the U.S. It is hoped shortly to be able to selectively blacken *tube-in-strip* which will permit very large flat plate *roof* collectors to be constructed.

††Calculated for a tilt of 45 deg air gap of 1-in. (plate to glass cover), convection coefficients slightly different from those given in Refs. 1 and 2 have been used.

The real advantage of the black body cover is that the small hole becomes an almost perfect black body, i.e. its absorptivity α to sunlight is almost unity (for light entering the hole has great difficulty in getting out), compared with the value of $\alpha = 0.90$ assumed for the blackened plate surface. This device for increasing the "blackness" of the collector is possible because one can afford, in this case, to forego the high concentration factor of 1840 and use a value of only 97. Such black "cavities" have been mentioned in the early solar energy patent literature and have been used recently by Trombe in his solar furnace work.

AUTHORS' CLOSURE (Mr. Nevins): Dr. Tabor's discussion of the paper is appreciated and I wish to express high regard of the authors for Dr. Tabor's comments. I would like to comment on two items.

In regard to the black body cover, the opening diameter of 1.375 in. was chosen to allow clearance between the metal cover and the image of the sun (approximately ¼ in. diameter), the hole being located in the same plane as the focus. During the tests reported here, the sun was tracked by hand since the automatic tracking devices had not been installed and this clearance was deemed necessary. Had the focus shifted to the metal cover, damage due to the resulting high temperature would occur. P, the concentrating power, is not affected by the black body cover if the present physical arrangement is used. The cover does, however, increase the effective absorptivity of the collector plate.

The second item in Dr. Tabor's discussion to which I would like to call attention is his comments on retention efficiency and his mention of *selective* surfaces for solar collectors. Analysis will show that the collection efficiency can be greatly improved if the surface has an absorptivity of 0.9 and an emissivity of 0.1. When these materials are made available in the United States, the efficiencies of flat plate collectors can be increased to values approaching those reported in this paper.

1624

No. 1624

PERFORMANCE OF A SOLAR HEATED OFFICE BUILDING

By F. H. Bridgers*, D. D. Paxton* and R. W. Haines†, Albuquerque, N. M.

WHAT is believed to be the first solar heated office building in the world, located in Albuquerque, N. M., was occupied in August 1956, and has operated successfully during the 1956-57 heating season. The 4,300 sq ft building has flat plate collectors, south facing, inclined 30 deg from the vertical, with single pane cover of ordinary double-strength glass, and a net collector area of 750 sq ft.

A 6,000-gal insulated underground water storage tank is used to store collected solar heat for cloudy weather and night-time use. A nominal 7½ ton package water chiller is used as a heat pump in winter and for cooling the building in summer. In winter, the water chiller is used as a heat pump only when the storage water temperature is not at a high enough level to satisfy the building heating requirements, and is automatically placed in operation by means of a space thermostat. An evaporative water cooler serves as a cooling tower for the heat pump on the summer cooling cycle. It is also used as a direct means of cooling the building in intermediate seasons. The distribution system within the building consists of a floor panel heating system in the high ceiling portion (north of the collector plates), a ceiling panel heating system in the low ceiling portion (south of the collector plates) and a forced air ventilation system serving all areas. The ventilating system has a 6-row water coil to warm air in winter and cool air in the summer. All items of equipment in the building are of standard manufacture with the exception of the collector plates. The collector plates consist of flat aluminum plates painted black with heat absorbing paint and ½ in. OD copper tubes soldered to form a continuous bond on the back side. Water passes through the copper tubes to remove the collected solar heat.

The building has structural steel bents supporting wood stud walls with 3 in. of glass fiber insulation. The solar collector plates are supported on a wood framework on the south side of the building.

The heat loss of the building based on 70 F indoors and 0 F outdoors is 118,570 Btu per hr, as calculated by the conventional method. This is equivalent to 29 Btu per hr heat loss per sq ft of floor area.

The total cost of the building was $58,500 or $13.60 per sq ft. The mechanical system, including the plumbing, heating and air conditioning was $17,400 or $4.05 per sq ft.

---

*Partner, Bridgers & Paxton, Consulting Engineers, Member of ASHAE.
†Chief Engineer, Bridgers & Paxton, Consulting Engineers, Member of ASHAE.
Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, January, 1958

Instrumentation for determining the performance consists of thermometer wells with test thermometers in the piping entering and leaving storage tank, solar collectors, heat pump evaporator and condenser, and heating coils. A separate 3-phase electric meter is provided for the heating system, independent of lights. A rotameter is located in the circulating water line to the collectors. All solar radiation data were obtained from the Weather Bureau which is about 3 miles from the building.

The system has been designed to provide year-around comfort at the lowest feasible operating cost. Controls are arranged to obtain 5 different water circuits to heat or cool the building depending on outside weather conditions. The direct

FIG. 1—SOLAR BUILDING, ALBUQUERQUE, N. M., SHOWING FLAT-PLATE COLLECTORS WHICH FORM MOST OF SOUTH SIDE

solar heating circuit is schematically illustrated by Fig. 2. Water is either pumped through the solar collector or bypasses the collector as determined by a differential thermostat with one bulb at the pump discharge and one bulb at the face of the collector plates. Whenever the temperature at the face of the collectors is lower than the entering water (such as it would be at night), the water bypasses the collectors. A room thermostat determines how much of the water passes through the heating system and how much bypasses to continue collecting heat. Whenever the storage water temperature drops below a level required to maintain the building temperature, such as after long periods of cloudy weather, the space thermostat automatically changes the water circuits as illustrated by Fig. 3 and starts the water chiller which is used as a heat pump. The heat pump continues to operate until the room thermostat is satisfied. After the room thermostat is satisfied, the water circuit will change back to direct solar heating (Fig. 2), provided the temperature of the water in the storage tank is not lower than the water circulating through the heating system. A differential thermostat with one bulb in the condenser pump discharge and the second in the water leaving the storage tank, will not allow water from the storage tank to circulate through the heating

system until the storage tank water is warmer than the water circulating through the heating system. It is possible to supply the full design heat output to the building with the heat pump even if the storage water temperature should drop to 40 F. Figs. 3, 4 and 5 illustrate various cooling cycles. In spring and fall, heating is required during night and morning. while cooling is required in the afternoon. Fig. 3 illustrates a cycle in which the evaporative water cooler is used to cool the building while at the same time solar heat is stored in the storage tank. During the early and late part of the cooling season, the evaporative water cooler is operated all night and the water that is cooled by the low wet-bulb night air is stored in the storage tank for use the following day. This cycle is illustrated by Fig. 5

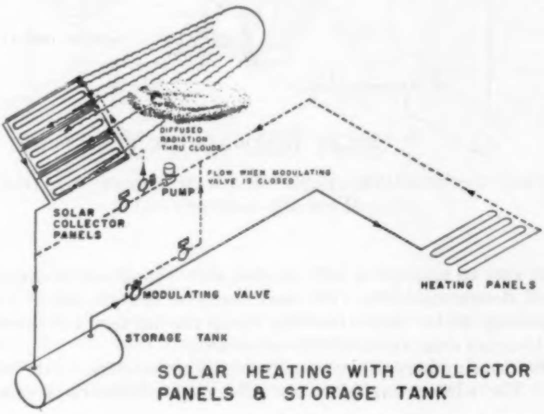


FIG. 2—SCHEMATIC FLOW DIAGRAM—DIRECT SOLAR HEATING

and provides for year-round utilization of the storage tank. When the cooling loads are too high to use the evaporative water cooler directly, it becomes a cooling tower for the packaged water chiller, and conventional chilled water cooling is accomplished as illustrated schematically by Fig. 6.

All changeovers between the various cycles described are accomplished by a manual selector switch except that the changeover between direct solar heating and the heat pump heating is automatic.

A few operating difficulties arose when the system was first started, such as corrosion causing pinhole leaks in the original aluminum collector plates with integral water passages (repaired by soldering copper tubes to the back of the aluminum plates), freeze-up of water in collector panels that did not drain properly, and inadequate venting due to the large quantities of air introduced when the collector plates were drained. The collector panels are drained automatically if panel temperature approaches the freezing point. After the necessary corrective measures were taken to rectify these difficulties, the system operated without any problems for the rest of the heating season.

Use of a heat pump with a solar collector has much to recommend it, particularly if cooling is a consideration. In cold and cloudy weather, the storage and collector

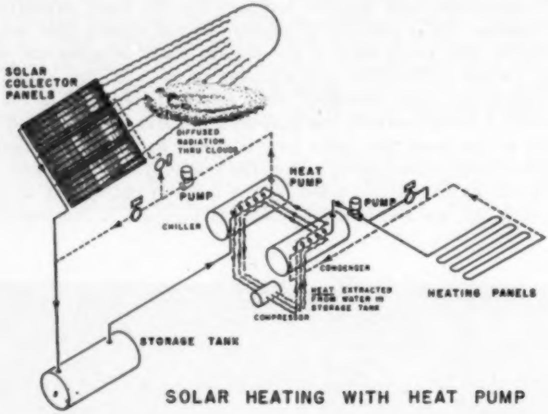


FIG. 3—SCHEMATIC FLOW DIAGRAM—SOLAR HEATING WITH HEAT PUMP

temperatures may be allowed to fall very low with a resultant increase in collector efficiency and storage capacity. The heat pump permits the use of a smaller collector and storage tank. Since the heat pump can be used for summer cooling, the system becomes more economically attractive.

Table 1 shows a summary of the sunshine and weather data for the 1956-57 heating season. The winter was relatively mild (4015 degree-days compared with

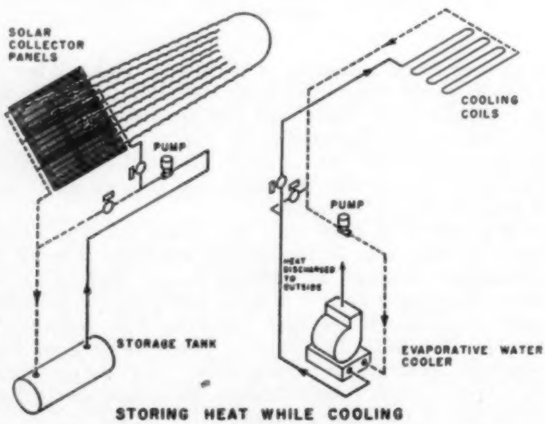


FIG. 4—SCHEMATIC FLOW DIAGRAM—SIMULTANEOUS HEAT COLLECTION AND COOLING

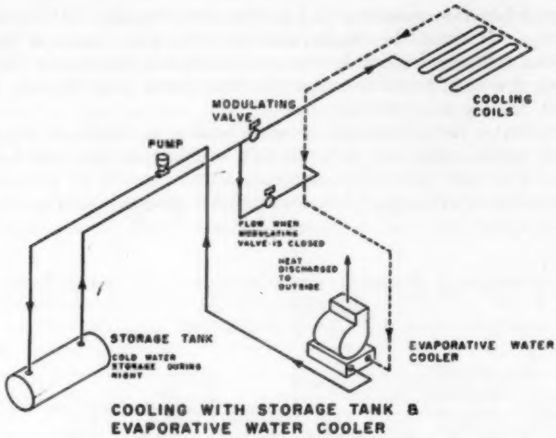


FIG. 5—SCHEMATIC FLOW DIAGRAM—COOLING WITH EVAPORATIVE WATER COOLER

4389 normal degree-days), however, January had more cloud cover than any other January in Weather Bureau history. Note in Table 1, column 5, that the percentage of possible sunshine was only 49 percent which was lower than New York City, Chicago, Boston or Philadelphia for the same period.

Table 2 summarizes the amount of heat pump operation. For the entire heating season, direct solar heating (Fig. 2) supplied 62.7 percent of the total heating

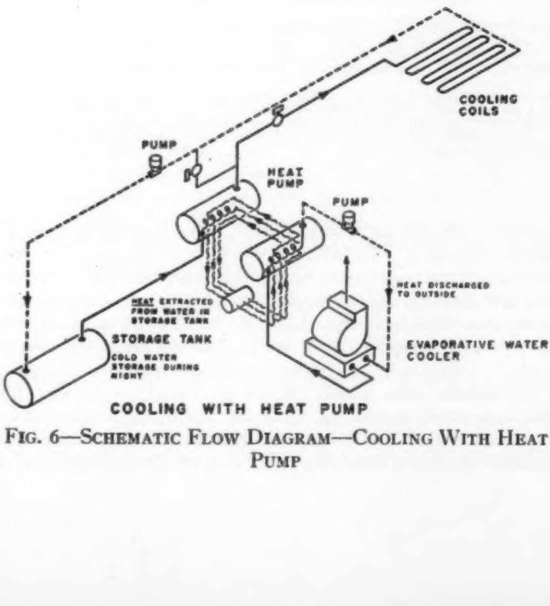


FIG. 6—SCHEMATIC FLOW DIAGRAM—COOLING WITH HEAT PUMP

requirements while the remaining 37.3 percent was supplied with the heat pump unit operating. It should be emphasized that the major source of the heat was from the solar collectors, which is true even when the heat pump was operated. The amount of energy required to run the heat pump supplied only 8.2 percent of the total heating requirements.

Fig. 7 illustrates the advantages of solar heating in regard to the problem of depletion of energy reserves. Solar heating in Albuquerque with a heat pump using steam-generated electricity consumes approximately 25 percent as much energy (excluding solar energy) as a conventional gas-fired heating system.

TABLE 1—SUMMARY OF SUNSHINE AND WEATHER DATA, SOLAR BUILDING, ALBUQUERQUE, N. M.

1	2[a]	3[b]	4[a]	5[a]	6[a]	7[c]	8[d]	9[e]
Month '56-57	Avg Temp, F deg		Degree Days	Possible Sunshine Percent	Avg Langleys, Gram Calories per sq cm	Avg Storage Tank Temp	Avg Collector Efficiency, Percent	Slope Factor
	Outdoor	Indoor						
October	60.3	—	195	83	468	—	—	—
November	41.0	73.0	715	91	367	87.2	59.5	2.2
December	36.3	71.4	880	85	306	80.1	60.0	2.3
January	39.9	72.0	771	49	258	59.2	66.0	2.2
February	48.1	73.5	468	63	390	77.2	59.0	2.1
March	47.3	73.2	543	64	488	83.5	52.0	2.0
April	54.2	—	317	73	620	88.6	54.0	1.8
May-June[f]	—	—	126	—	—	—	—	—

[a]Columns 2, 4, 5 and 6 taken from Weather Bureau Data as summarized in Reference 1. Langleys are measured on a horizontal surface.

[b]Column 3 is a 24-hr average indoor temperature as recorded in two different parts of building 3 times per day.

[c]Column 7 is an average storage tank temperature of water leaving tank as recorded 3 times per day.

[d]Column 8 is calculated using the outdoor temperature (Column 2) and average collector surface temperature (Column 7 plus 8 F) with curves in Reference 5. For November and December curves for a *clear day* were used, and for January, February, March and April, curves for partly cloudy days were used.

[e]Column 9 is the experimental ratio of incident radiation for a collector at optimum slope to the incident radiation on horizontal surface based on the work of Bliss[2] to determine the factor of 2.3 for December. Other values are calculated by geometric relationships as given by Hottel[9].

[f]Cooling was done considerably more in May and June than heating so that degree-days is the only pertinent data for these months.

Fig. 8 illustrates the heating energy cost of solar heating compared with other conventional type systems. The energy cost for the Albuquerque Solar Building was $78.46 (the electrical cost for the heat pump operation was $0.02 per K.W.Hr.) where the energy cost for a conventional gas-heating system would have been $162.90 plus $6.00 minimum charge making a total of $168.90. (The charge for small gas quantities averages $0.70 per MCF where for large uses it would decrease as low as $0.44). This is a 53 percent saving in fuel cost for a winter with a lower percentage of sunshine than usual. However, based on present fuel prices, a savings of this magnitude would not make the additional initial cost economically sound. The trend of fuel cost increasing at a faster rate than electrical energy cost will probably continue, particularly in view of the possibility of atomic gen-

erated electrical energy. This could make a heat-pump solar-heated system economically attractive in a relatively short time.

Fig. 9 illustrates the relationship between the heat pump operation and the most important variables that determine how often the heat pump will operate, such as the number of degree-days and the solar radiation intensity. Note that a large change in degree-days or solar radiation is not followed by a change in heat pump operation until a week later. This is due to the large capacity of the storage tank.

Fig. 10 shows a comparison of the calculated storage tank temperature and the recorded temperature. The method of calculation has been improved from that

TABLE 2—SUMMARY OF HEAT PUMP OPERATION, 1956-57, SOLAR BUILDING, ALBUQUERQUE, N. M.[a]

MONTH	TOTAL HEATING REQUIREMENTS MBTU	HOURS OF HEAT PUMP OPERATION, PER MONTH	HEAT TO BUILDING WITH HEAT PUMP OPERATING, MBTU	RATIO OF HEATING WITH HEAT PUMP TO TOTAL HEATING
October	7,900	0	0	0
November	29,100	33.1	4,440	0.152
December	35,700	111.3	14,500	0.406
January	31,300	216.0	25,300	0.810
February	19,000	75.4	9,260	0.487
March	22,000	39.3	5,260	0.239
April	12,900	14.8	1,980	0.154
May-June	5,000	0	0	0
Total for Heating Season	162,900	489.9	60,740	0.373

[a]Total energy supplied to operate heat pump—13,350 MBtu (3,923 K. W. Hr.); average coefficient of performance of heat pump—4.5; total energy for heating from collectors (heat pump and direct solar) —149,550 MBtu (91.8 percent).

Note. Total heating requirements calculated from degree-days from Reference 1. The total for the season includes degree-days for October and May.

described in a previous paper[4] and is illustrated in the Appendix. For the comparison shown in Fig. 10, the heating required was calculated by the conventional degree-day method (with a 65 F base). The slope factor (2.3 for December 21) used for the ratio of incident radiation on a collector at optimum slope to that on a horizontal collector was applied only the percentage of time that the sun was shining, since collection of diffuse radiation is independent of slope. The disagreement toward the end of the month can not be readily explained, although some of the reasons may be as follows:

1. The calculations are based on U. S. Weather Bureau data which is recorded at a site about 3 miles from the Solar Building. On partly cloudy days, cloud conditions may vary considerably from one area to the other.

2. Water circulation in the solar collectors could have been temporarily stopped due to air in lines.

Fig. 11 is an attempt to show some correlation between the factors affecting the ratio of the heat which must be supplied by an auxiliary heating unit to the total

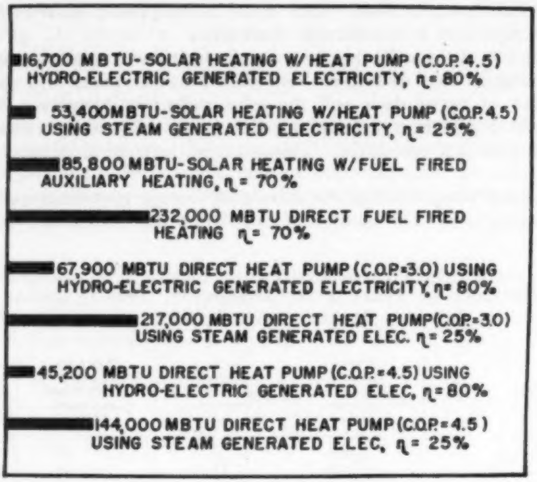


FIG. 7—COMPARATIVE DEPLETION OF ENERGY RESERVES FOR DIFFERENT TYPES OF HEATING SYSTEMS BASED ON PERFORMANCE OF SOLAR BUILDING (1956–57)

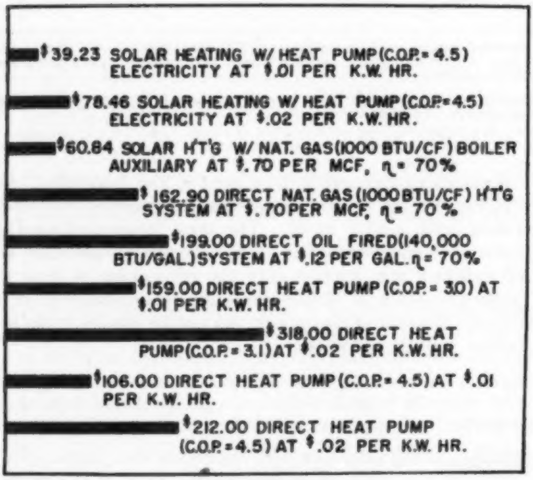


FIG. 8—COMPARATIVE OPERATING COSTS FOR DIFFERENT TYPES OF HEATING SYSTEMS BASED ON PERFORMANCE OF SOLAR BUILDING (1956-57)

heat requirements. The abscissa is a parameter containing the factors that have the most important effects on the collection of solar heat, and the total heat requirements. (See Appendix for Nomenclature of symbols in the parameter). The point for February is far off the curve, which can be logically explained by the fact that the storage water temperature was very low after an unusually cloudy January, and although the weather for February was mild and had a comparatively large amount of solar radiation intensity, the heat pump had to operate almost continually until the storage temperature became high enough to heat the building directly. It is readily admitted that there are not enough points to make the curve shown on Fig. 11 valid, but if such a curve were valid, it would be possible to predict fairly quickly the economic feasibility of a solar heating system for any locality. All variables in the parameter can be obtained directly from Weather Bureau Data (Reference 1) except efficiency and slope factor which can be calculated as outlined in References 3 and 5. A family of curves like the one shown in Fig. 11, representing various ratios of Btu per hr heat loss per sq ft of collection could be prepared to allow a quick economic study with a large variation in design

NOMENCLATURE

CF = cubic foot.
C.O.P. = coefficient of performance.
DD = degree-days (base 65 deg).
K.W.Hr. = kilowatt hour.
L = solar radiation in Langleys (gram-calorie per sq cm per day).
MBD = thousand British thermal units per day.
MBtu = thousand British thermal units.
MCF = thousand cubic feet.
SF = slope factor.
%SS = percent of possible sunshine.
η = efficiency.
η_c = efficiency of solar collector.

conditions. The points on the curve of Fig. 11 will be carefully checked during the coming heating season.

It is believed that the test data are reasonably accurate for field type test conditions. The solar radiation data could be greatly improved by having a pyrheliometer at the building rather than depending on Weather Bureau data recorded 3 miles away.

It is believed that the improved design procedure as illustrated in the Appendix is satisfactory for determining the range of water temperature to be encountered under the most adverse conditions. As shown by Fig. 10, the actual storage water temperature appears to change more slowly than calculations would indicate which is probably due to heat entering and leaving storage in the building structure.

It is felt that a water system with a heat pump is a very satisfactory system as far as giving year-round comfort and ease of operation.

Before an overall economic evaluation can be made of the building, the operating cost data for the cooling season will have to be included. At this time, a full cooling season has not been completed. With the present cost of fuels for heating, the savings in fuel costs do not justify the necessary first cost expense for a solar heating system in most localities in the United States. However, there are some areas where the high fuel cost would make a solar heating system economically

sound. With the continual depletion of fuel reserves— the increasing research activity in the solar energy field, the increasing demand for air conditioning, and the possibility of future cost reduction in solar collectors, the solar heating system with a heat pump may be economically attractive over a large part of the United States in a relatively short time.

REFERENCES

1. Climatological Data—National Summary (U. S. Dept. of Commerce, Weather Bureau, Monthly Publications, June, 1956 to April, 1957).

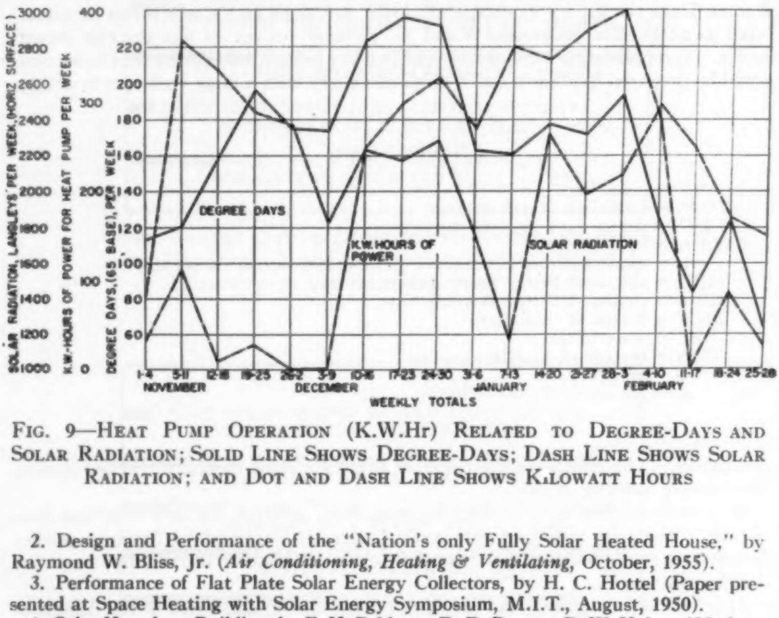


FIG. 9—HEAT PUMP OPERATION (K.W.Hr) RELATED TO DEGREE-DAYS AND SOLAR RADIATION; SOLID LINE SHOWS DEGREE-DAYS; DASH LINE SHOWS SOLAR RADIATION; AND DOT AND DASH LINE SHOWS KILOWATT HOURS

2. Design and Performance of the "Nation's only Fully Solar Heated House," by Raymond W. Bliss, Jr. (*Air Conditioning, Heating & Ventilating*, October, 1955).

3. Performance of Flat Plate Solar Energy Collectors, by H. C. Hottel (Paper presented at Space Heating with Solar Energy Symposium, M.I.T., August, 1950).

4. Solar Heat for a Building, by F. H. Bridgers, D. D. Paxton, R. W. Haines (*Mechanical Engineering*, June, 1957).

5. ASHVE RESEARCH REPORT 1502—Availability and Utilization of Solar Energy Parts I, II and III, by R. C. Jordan and J. L. Threlkeld (ASHVE TRANSACTIONS, Vol. 60, 1954, pp. 177-238).

APPENDIX

The equations used in the calculations are derived as follows:

For the heat required: The heat loss was calculated at 118,570 Btu per hr including ventilation air, based on 0 F outside and 70 F inside. Heating requirements were based on the conventional degree-day method.

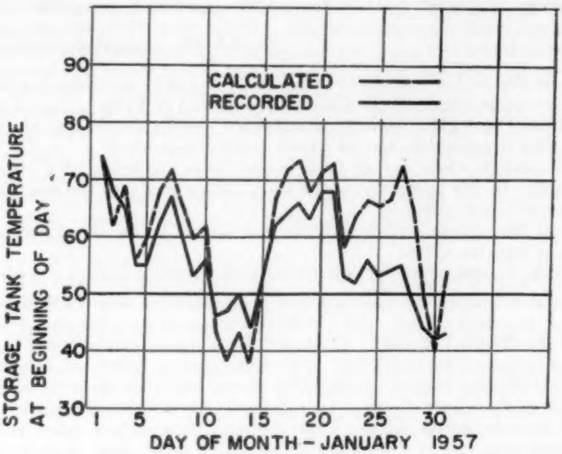


FIG. 10—COMPARISON BETWEEN CALCULATED STORAGE TEMPERATURE AND RECORDED STORAGE TEMPERATURE

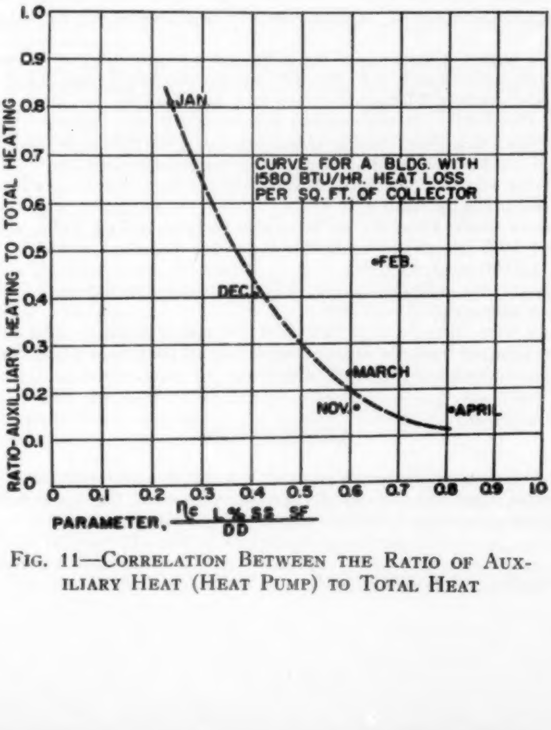


FIG. 11—CORRELATION BETWEEN THE RATIO OF AUXILIARY HEAT (HEAT PUMP) TO TOTAL HEAT

118,570 Btu/hr. $\times$ 1/1000 $\times$ 24 hr day $\times$ (65 − avg daily temp)/70

= 40.6 (65 − avg daily temp), MBD ..(1)

Although the degree-day method is based on 70 F inside, no correction was included for the higher inside temperatures maintained as shown in Table 1, since it was assumed that the internal heat gains from lights and other sources would offset the theoretically greater heating requirements for the higher inside temperature.

For those periods when storage tank temperatures are below 85 F, the heat pump must be used. In this case some of the work energy goes into the condenser water as heat and must be accounted for.

The actual solar energy withdrawn from storage will be the calculated heat required multiplied by a factor of 0.8.

The average temperatures are determined from Weather Bureau records.

For the heat collection: The heat collected is a function of solar radiation availability, efficiency, and collector area. Solar radiation received on a horizontal surface is recorded by the Weather Bureau. However, this must be extrapolated to the sloping, southfacing surface of this collector. This extrapolation factor can only be estimated. The Weather Bureau Station at Blue Hill records radiation data on vertical surfaces facing each of the cardinal points as well as on the horizontal surface. Jordan and Threlkeld[5] have developed curves for extrapolating these data to other latitudes.

The values also can be calculated theoretically using solar altitude and incident angle. We actually used a multiplier of 2.3 for the slope factor for January and December, which was based on experimental work by Bliss[2]. It is hoped that a pyrheliometer can be obtained and mounted on the sloping collector to confirm this. The multiplier of 2.3 would be applicable only for periods when the sun is shining since collection of diffuse radiation on cloudy days would not be affected by the slope of the collector.

Weather Bureau data are given in Langleys per day. One Langley is one gram-calorie per square centimeter. A multiplier of 3.69 converts this to Btu per day per square foot.

Efficiency is largely a function of the difference between collector temperature and outdoor temperature, though it is also affected by cloudiness, particularly at greater temperature differentials. Efficiencies used here are based on the curves developed by Jordan and Threlkeld[5]. In determining efficiencies, it was assumed that collector temperature was 8 deg above storage temperature and the difference used was that between average outdoor temperature and average collector temperature. This required a trial-and-error solution since final collector temperature is not known until calculations have been made for the day.

Then, *collection equals:* Langleys (on horizontal surface) $\times$ 3.69 $\times$ 2.3 $\times$ 750 sq ft $\times$ efficiency $\times$ 1/1000 (to convert to MBD) $\times$ %SS + Langleys $\times$ 3.69 $\times$ 750 sq ft $\times$ efficiency $\times$ 1/1000 $\times$ (1 − %SS).

Or, *combining terms, collection equals:* (6.36 $\times$ Langleys $\times$ efficiency) (%SS) + (2.76 $\times$ Langleys $\times$ efficiency) (1 − %SS).

Calculations were then made to determine use and collection. The difference was added to or subtracted from the storage temperature at the rate of 50,000 Btu per degree storage temperature change (6000 gal of water).

DISCUSSION

H. TABOR*, Jerusalem, Israel (WRITTEN): I would like to congratulate the authors on a very useful paper and in particular for the inclusion of Fig. 7 which discusses the depletion of energy reserves for different types of building.

*The National Physical Laboratory of Israel.

G. O. G. Lof in the March 1957 issue of *The Sun at Work* has criticised a former paper by Mr. Haines describing the same building, wherein no account was taken of the heat-pump energy, for it is not sufficiently well recognized that a Btu added to a system by an electrically-operated heat pump has cost 3 to 4 Btu at the fuel-operated power station and that an electrical Btu may cost 5 to 10 times a fuel Btu.

The present paper puts this right and shows the real significance of solar heated houses (with heat-pump assistance) as savers of fossil fuels. Thus, from Fig. 7, using steam generated electricity (η = 25 percent) only 53,400 MBtu are consumed compared with 232,000 MBtu for direct fuel-fired heating (η = 70 percent) or a saving of 77 percent of fuel. If there is any national (or international) policy on fossil fuel conservation, then an excellent case is made for the solar heated house even if its superficial economics (to the owner) are not yet attractive.

In this connection, would the authors care to indicate how the cost of $17,400 for *mechanical system including plumbing heating and air-conditioning* is made up and what this figure would have been if the same building had been built:

(a) without solar collectors but with the heat pumps using underground or well-water heat source,

(b) with conventional fuel-fired boiler—no solar energy and no heat pump,

(c) with conventional fuel-fired boiler for winter and cooling unit (reversed heat pump) for summer?

D. T. DONOVAN, Tyler, Texas (WRITTEN): This paper presents the results of a practical working laboratory for the theories put forth in the papers of Professors Jordan and Threlkeld at the Houston meeting in 1954. The unique manner in which the solar collector, the heat pump and the evaporative water cooler together with the attendant control system were combined is most ingenious and interesting.

The paper aroused several questions as follows.

(a) At what depth was the underground storage tank located? What are the ground temperatures at this depth?

(b) Did the authors make a direct cost comparison between the cost of the installed heating and air-conditioning system and a conventional heating and cooling installation with hot water or steam heating and water-cooled or air-cooled cooling equipment?

(c) In the cost comparison charts, Figs. 7 and 8, efficiency values for gas and oil systems are given as 70 percent. *A. G. A.* standards for gas furnaces and industry practice for oil furnaces show output as 80 percent of input.

We look forward to receiving additional data from the authors on the operation of this installation during the cooling season.

H. H. REICH, Pittsburgh, Pa.: I believe my question really applies to both the papers by Mr. Davis and by Mr. Bridgers, but maybe Mr. Bridgers has the information more readily available. The question is: How is the calculated heat loss of 100,000 Btu per hr compared with the actual heat loss?

Then there is also a second, more practical question, namely: How did the heat loss through the south wall enter into the calculations since apparently there is heat absorption by the wall, and yet there would also be a heat loss through the wall? Then too, the peculiar construction of the wall in the paper by Davis and Lipper would seem to put heavy emphasis on heat transmission calculations.

AUTHORS' CLOSURE (Mr. Bridgers): We appreciate Professor Tabor's remarks. Professor Tabor is one of the world's outstanding scientists in solar energy research. He asked for a breakdown of the cost of the mechanical system. The breakdown was approximately as follows: The cost of equipment, including heat pump, automatic controls, solar collector panels, was $7591. The cost of the ventilation ducts and grilles was $1244. The cost of the heating piping and installation of equipment connected

with the solar panels was $6388. The cost of the plumbing was approximately $1600. The insulation for the tank and ducts was $1177. Making a total of $17,400.

I might add that while every piece of equipment was purchased, being in the business of designing heating and air-conditioning equipment, I think it should be brought out that we got prices lower than the average buyer could. I would estimate that, if the man on the street were to buy the same building and equipment, the cost would be from 10 to 30 percent higher.

The next question asked by Professor Tabor was: What would have been the figures without solar collectors but with the heat pump using underground wells for water source heat?

In this case the solar collectors would have to be replaced by a deep well and the location of the ground water being about 200 feet below the surface would require a 300-foot deep well.

Just on an estimated basis it is felt that the well would cost approximately $3000 and, if a conventional wall would replace the solar collector panels, it would be about a $1200 deduction in the cost of the building, which makes approximately a net addition of $400. In other words, a system with a heat pump and a deep well for this particular building at the location shown would have cost approximately $400 more than the solar-heated system.

Then he wanted to know the cost with conventional fuel-fired boiler and no solar energy or no heat pump. The estimated net deduction would be $5400 or 31 percent. However, the same things are not being compared. You are comparing a straight heating system with a year-round heating and cooling system for this particular question.

The next question was: With a conventional fuel-fired boiler for winter and a cooling unit for summer, our estimate of the net deduction would be $3000 or 17 percent.

All of these figures or estimates are based on the same type of heat and cooling distribution system, which was fairly deluxe in this particular building. There were floor panels for heating the building in the high bay area and ceiling panels for heating the building and cooling the building in the low bay area, a ventilation air system that served the entire building with heating and cooling coils and year-round controls.

In answer to Mr. Donovan's question, he asked what the depth of the tank was. The top of the tank is approximately 2 feet below the surface. It is estimated that the average ground temperature would be close to 60 degrees, although this was not actually measured.

Again he asks the comparison between a conventional system, the comparative cost between a conventional system and the solar collector system. We would estimate a net deduction of $3000 for conventional system having the same type of heating and cooling distribution system.

He mentioned that the *A.G.A.* listed boiler efficiencies as 80 percent. Our thoughts here in using 70 percent is that the 80 percent is with the burners adjusted theoretically perfect and we thought that a more realistic figure over a year's operation with normal maintenance and off-on operation would be an average figure of 70 percent.

Mr. Reich asked how does the calculated heat loss compare with the actual heat loss. Actually we didn't undertake to prove or disprove the ASHAE method of calculating heat loss and we did no work in that regard, although we know that it is very close. Based on the capacity of the heat pump unit, we know that the comparison is quite close, but I am not prepared to give an exact deviation.

He asked how the heat loss from the south wall enters into the problem. Actually, you have to make a heat loss on the south wall just as you do other walls because at night when solar energy is not collected, it is still necessary to heat the building and the over-all transfer coefficient through the collector wall must be included in the heat loss calculation. Actually it is quite a low number since the collector panels have 4 inches of insulation behind them.

1625

No. 1625

SUN ENERGY ASSISTANCE FOR AIR-TYPE HEAT PUMPS†

By Chester P. Davis, Jr.* and Ralph I. Lipper**, Manhattan, Kans.

THE AIR-TYPE heat pump has become the major type of heat-pump unit manufactured. These units, using air as a heat source, are influenced more than other types by the extremes, the severity, and the variability of outdoor air temperature and relative humidity. Low outdoor temperature decreases heating capacity and coefficient of performance (CP) during the period when the greatest demand for heat is required. Stated another way, the heat loss of the conditioned space is usually maximum when the heat source for the heat pump is the poorest from a thermodynamic viewpoint. Defrosting problems of the outdoor coil may be serious during high humidity periods when outdoor air temperatures are 28 to 38 F.

These limitations have delayed and limited acceptance of this type of year-round air-conditioning equipment in northerly areas where more severe winter climatic conditions are encountered. The decreased efficiency and capacity in these areas may require appreciable quantities of supplemental electrical resistance heat. Extensive usage of these heaters results in poorer demand and load factors than is experienced with this equipment in milder climates. These factors offer less incentive to the electric utility to provide a more favorable electric rate structure for home heating with the heat pump.

Limited studies[1] indicate that the availability of solar radiation is above average during periods of extremely cold weather. The principal exceptions are in the vicinity of the Great Lakes and other large bodies of water.

It was with these factors in mind that heat pump research attention at Kansas State College was turned to utilization of sun energy as a logical supplement to an air-source heat pump system in late 1953 and 1954. At that time, a considerable

---

†This paper was approved as *Scientific Journal Article No.* 81, Kansas Agricultural Experiment Station.
*Research Engineer, Farm Electrification Laboratory, Agricultural Engineering Research Division, Agricultural Research Service, U.S. Department of Agriculture, in cooperation with the Kansas Committee on the Relation of Electricity to Agriculture and the Kansas State College Agricultural Experiment Station.
**Associate Professor, Agricultural Engineering Department, Kansas State College.
[1]Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, January 1958.

quantity of basic information for system design had become available from theoretical studies at the University of Minnesota.

PLAN FOR UTILIZING SOLAR ENERGY

After a search of the literature pertaining to solar collection and with several practical considerations in mind, it was decided to study a simple low-cost flat-plate type collector for air heating. The collector visualized could serve a structural purpose in most applications. Air drawn through the collector would be the heat source for a heat pump. This air would be warmed during periods of solar radiation. It was anticipated that solar radiation received in excess of need during daytime hours might be stored for nighttime use.

By collecting and storing solar energy at low temperatures for delivery to the heated space via the heat pump cycle, losses from a simple collector and low-cost storage would be minimized. Direct use of solar energy for space heating was not contemplated in this study. However, such a low-cost collector appears to possess potential value for air warming either with or without the heat pump in certain agricultural applications. Many farm structures use the type of construction basically represented by this collector. Moderately heated air could be useful for grain drying or for animal-shelter ventilation.

COLLECTOR DESIGN

An adaptation was made of a typical insulated wall or roof surface covered with corrugated sheet metal painted black. This metal was fastened over sheathing nailed to studding or rafters. A five mil thick plastic film weatherable polyester was framed and placed with an intervening air space over the sheet metal. Function of the film was to pass solar radiation but trap long-wave re-radiation from the blackened plate. It was found desirable to limit deflection and billowing of the plastic by placing it in a sandwich of wire netting. This prevented the film from contacting the collector surface and conducting away significant quantities of absorbed heat. The air channel above the plate combined with the channel formed between the sheet metal and sheathing provided heat exchange from both surfaces of the sheet-metal absorber to the air forced through the collector. The corrugations were believed to increase that heat exchange, provide a larger area for absorption of solar energy and perhaps reduce reflection over that of a flat-plate surface. No attempt has been made to confirm how significant this geometry is in terms of increased gain in heat collection.

Heat transfer due to the log-mean temperature difference between the collector plate and air passing through it was known to be greater for shorter travel distance through the collector. Using temperatures of collector-heated air, this is shown for 2 radiation levels in Fig. 1. Since these data were taken in late May, the final temperature of the air was considerably higher than realized during winter operation though the temperature rise from the inlet condition was comparable. On the basis of this information, it was found desirable to introduce air from both ends of the collector and exhaust it from a plenum located at the center of the collector panel.

Several plastic films of the polyethylene and vinyl type were studied but exposure over a period of a few months brought progressive deterioration, reduced solar

transmission, and ultimate failure. This was due to ultraviolet attack on the plasticizers used in formulation.

As in other solar research there was some question as to the respective merits of different orientations of the collector surface. After consultation with architects and study of the significance of departure from optimum it was initially decided that vertical orientation would be used for these studies. The considerations influencing this decision were that the collector could be satisfactorily in-

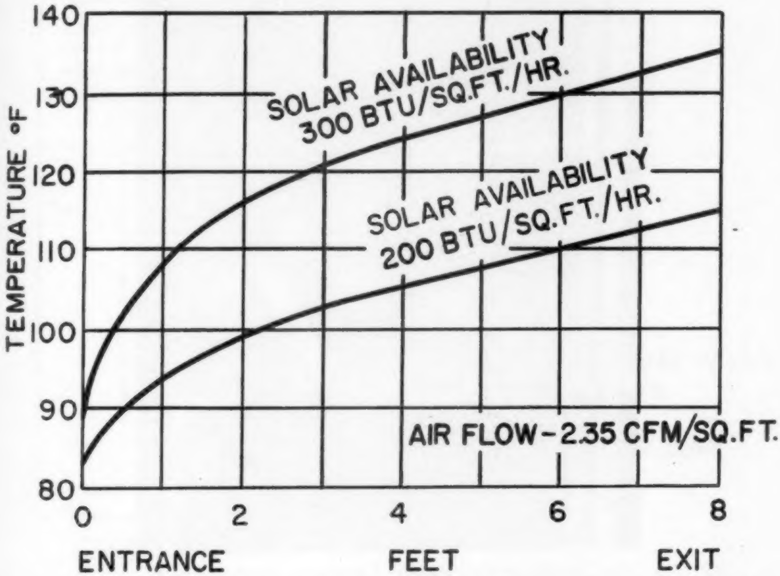


FIG. 1—TEMPERATURE RISE IN TRAVEL THROUGH COLLECTOR AT 2 LEVELS OF SOLAR RADIATION AVAILABILITY

tegrated as a portion of the house wall structure, an extension of that surface, or a yard screen. Other practical considerations are ease of repair, reduction of hail damage, and adaptability to shading. Shade from an overhanging roof or shielding panels would reduce the absorption of summer solar-heat load. The chief advantage to an optimum inclined orientation is the absorption of a larger proportion of available solar radiation. For the period of maximum need for supplemental heat (November 1 to March 1), the reduced collection of solar radiation by using a vertical over an optimumly oriented surface is of the order of magnitude of 5-15 percent. Average reduction is less than 10 percent for the latitudes under consideration.

An area of 160 sq ft of this type collector surface was provided (Fig. 2) for the scale model system (approximately one-fifth) for laboratory-type tests.

Preliminary operational data on the collector alone showed collector plate temperatures of 140 to 145 F at noon on bright winter days with outside temperatures

Fig. 2—Plastic-Covered Vertical, South Facing, Plate-Type Solar Collector Using Air-Heat Exchange

at 40 to 50 F. It was predicted from available radiation and this temperature difference[3] that the maximum collector efficiency would approach 45 percent. Data taken over a period of several days indicated actual average efficiency between 30 and 35 percent.

Laboratory Heat-Pump Systems

Two scale-model (¾ hp) independent systems were devised. One used outside air alone; the other used air after it passed through the collector (Fig. 3). Air could be diverted from one system to the other at any time to determine the extent

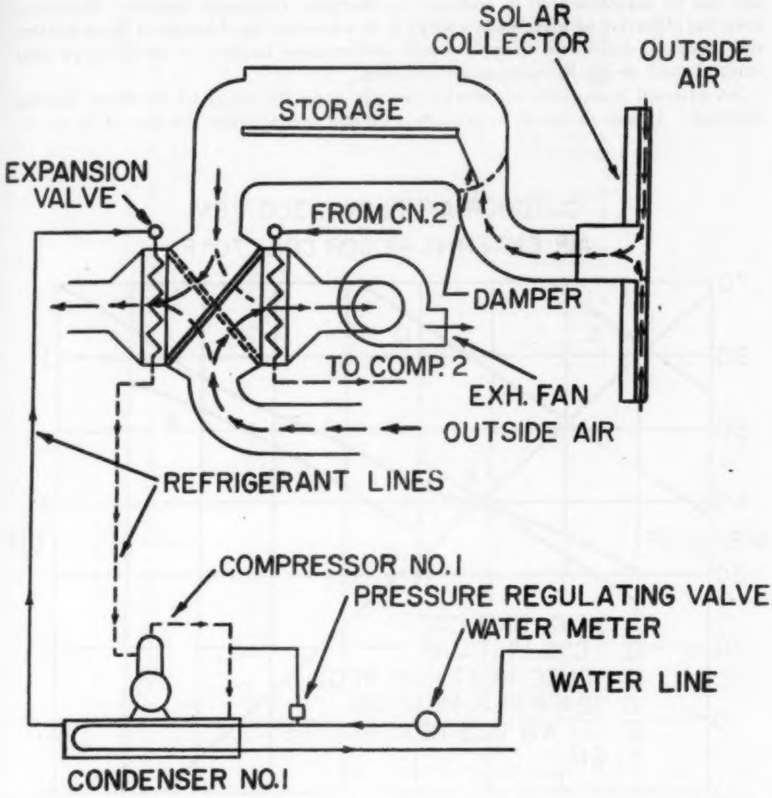


FIG. 3—SCHEMATIC OF AIR SOURCE AND SOLAR SUPPLEMENTED HEAT-PUMP SYSTEMS WITH STORAGE

of variance in system equipment performance. An initial series of tests was conducted using the collector and heat pump systems without heat storage.

STORAGE

During the peak portion of the solar day it was frequently noted that an excess of heat was collected in relation to that which could be used efficiently by the heat pump during those hours. As a result, air was discharged from the evaporator of the solar supplemented system at temperatures exceeding that of the outside air. This proved the need for storage to act as a *flywheel* or reservoir.

The concept adopted for storage requirements is that the collector and storage are supplemental, allowing considerable design freedom. Air is the primary source

and can be supplemented in addition by electrical resistance heaters. However, since the objective of using solar energy is to minimize the demand of these heaters and maintain satisfactory capacity and performance factors for the air-type heat pump, broad design limitations are imposed.

No attempt was made to provide storage capacity equal to the total heating required. It was designed to provide a 20-deg temperature rise for 24 hr to the

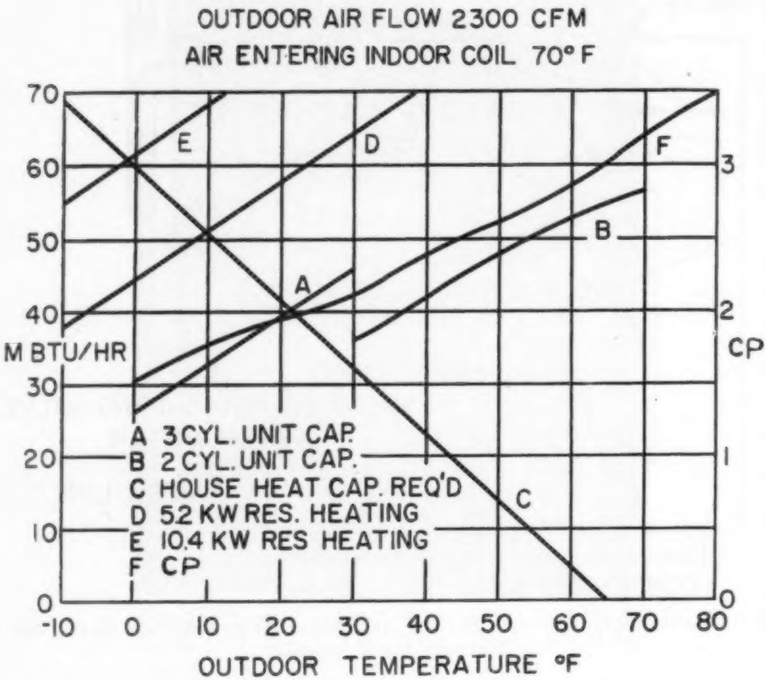


FIG. 4—PERFORMANCE CHARACTERISTICS OF COMMERCIALLY PRODUCED AIR-TYPE HEAT PUMP USED AS BASIS FOR HYPOTHETICAL ANALYSIS

air supply for heat pumps of current design. Heat transfer rates to and from storage were important considerations.

The *low side* (evaporator) storage designed and studied consisted of water placed in plastic-film-sealed tin cans 4½ in. in diameter and 7 in. high. Heat was stored or released in latent (at 32 F) and sensible form by the air en route to the heat pump. Cans were placed in overlapping rows in the storage bin so that the air made effective contact to provide heat transfer from or to the water. It was found necessary to incorporate an expandable, anti-freeze-filled sleeve along the

TABLE 1—HEATING REQUIREMENTS OF A HYPOTHETICAL HOUSE USING A HEAT PUMP OVER THE RANGE OF —7.5 TO 62.5 F OUTSIDE TEMPERATURE AND WITH 10- AND 20-DEG AVERAGE RISE THROUGH SOLAR COLLECTOR AND/OR STORAGE

OUTSIDE AIR TEMPERATURE	AIR TEMP. FROM COLLECTOR AND STORAGE		CP			HEAT PUMP CAPACITY (THOUSAND BTU)			REQUIRED CAPACITY (THOUSAND BTU)	ELECTRICAL HEATER CAPACITY			HEATER KWHR REQUIRED			HEAT PUMP KWHR REQUIRED			FAN KWHR	TOTAL KWHR		
T[a]	10° RISE	20° RISE	@ T	10° RISE	20° RISE	@ T	10° RISE	20° RISE	OUT-SIDE TEMP.	@ T	10° RISE	20° RISE	@ T	10° RISE	20° RISE	@ T	10° RISE	20° RISE		@ T	10° RISE	20° RISE
—7.5	2.5	12.5	1.32	1.58	1.82	21.5	28.0	34.6	67.0	45.5	39.0	32.4	13.33	11.4	9.5	4.78	5.2	5.58	0.28	18.11	16.88	15.36
—2.5	7.5	17.5	1.45	1.72	1.92	25.0	31.3	38.0	62.3	37.3	31.0	24.3	10.9	9.1	7.1	5.05	5.35	5.8	0.28	15.95	14.73	13.18
2.5	12.5	22.5	1.58	1.82	2.00	28.0	34.6	41.0	57.7	29.7	23.1	16.7	8.7	6.77	4.9	5.2	5.58	6.0	0.28	13.9	12.63	11.18
7.5[b]	17.5	27.5	1.72	1.92	2.08	31.3	38.0	44.5	53.0	21.7	15.0	8.5	6.35	4.4	2.5	5.33	5.8	6.3	0.28	11.68	10.48	9.08
12.5	22.5	32.5	1.82	2.00	2.18	34.6	41.0	37.5	48.5	13.9	7.5	11.0	4.07	2.2	3.2	5.57	6.0	5.05	0.28	9.64	8.48	8.53
17.5	27.5	37.5	1.92	2.08	2.32	38.0	44.0	40.5	44.0	6.0	0	3.5	1.76	0	1.02	5.8	6.2	5.1	0.28	7.56	6.48	6.4
22.5	32.5	42.5	2.00	2.18	2.46	39.3	37.5	39.3	39.3	0	1.8	0	0	0.5	0	5.75	5.05	4.7	0.28	5.75	5.38	4.98
27.5	37.5	47.5	2.08	2.32	2.57	34.8	34.8	34.8	34.8		0			0		4.9	4.4	4.0	0.28	4.9	4.68	4.28
32.5	42.5	52.5	2.18	2.46	2.68	30.0	30.0	30.0	30.0							4.03	3.58	3.28	0.28	4.03	3.96	3.56
37.5	47.5	57.5	2.32	2.57	2.82	25.5	25.5	25.5	25.5							3.22	2.91	2.76	0.28	3.22	3.19	3.04
42.5	52.5	62.5	2.46	2.68	2.95	21.6	21.0	21.0	21.0							2.5	2.3	2.09	0.28	2.5	2.58	2.37
47.5	57.5	67.5	2.57	2.82	3.12	16.5	16.5	16.5	16.5							1.88	1.72	1.55	0.28	1.88	2.00	1.83
52.5	62.5	72.5	2.68	2.95	3.26	11.8	11.8	11.8	11.8							1.29	1.17	1.06	0.28	1.29	1.45	1.34
57.5	67.5	77.5	2.82	3.12	3.45	7.2	7.2	7.2	7.2							0.75	0.68	0.61	0.28	0.75	0.96	0.89
62.5	72.5	82.5	2.95	3.26	3.60	2.5	2.5	2.5	2.5							0.25	2.2	0.2	0.28	0.25	0.50	0.48

[a]Temperature in Fahrenheit degrees. [b]See Appendix for sample calculation.

central axis within the cans and extending above the water surface to prevent can rupture. When the water froze, the anti-freeze solution pushed out and expanded in the container above the ice surface.

It was noted that some material which changes state at 45 to 55 F would be more satisfactory. Of several compounds used thus far, 1.2 dibromoethane has been the most satisfactory except for cost. To date, completely satisfactory, stable, and suitable compounds have not been found. Other studies in this field have sought

Table 2—Degree Hours Based on TAC of 5 F

	Hours	Multiplier Degrees	Degree Hours
$0<t<5$	50-0=50	62.5	3,125
$5<t<10$	100-50=50	57.5	2,875
$10<t<15$	250-100=150	52.5	7,875
$15<t<20$	450-250=200	47.5	9,500
$20<t<25$	775-450=325	42.5	13,813
$25<t<30$	1200-775=425	37.5	15,938
$30<t<35$	1775-1200=575	32.5	18,688
$35<t<40$	2550-1775=775	27.5	21,313
$40<t<45$	3350-2550=800	22.5	18,000
$45<t<50$	4050-3350=700	17.5	12,250
$50<t<55$	4675-4050=625	12.5	7,813
$55<t<60$	5400-4675=725	7.5	5,438
$60<t<65$	6100-5400=700	2.5	1,750

a material primarily intended for use at a higher temperature level and for a basically different application.

Instrumentation

Instrumentation was provided to measure refrigerant flow, pressures, and temperatures for state-condition determinaton. Provision was made for measurements of water flow and temperature change through the water-cooled condenser, for air flow, and for determining temperature outdoors, after the collector, at various points in the heat storage, and before and after the heat exchangers. Solar radiation and all other measurements except air flow were made with recording type instruments to provide a record for system analysis.

Method of Analysis of the Value of Solar Collection Combined with Air-Source Heat Pump

Prior to initiation of the past year's research, a study of the heating requirement for a hypothetical house was made using performance curves (Fig. 4) of a commercially produced air-type heat pump. A rational approach was thus sought to the analysis of the energy economy that might be realized through solar energy supplementation of that air source. An air flow of 1.5 to 3.0 cfm per sq ft as

mathematically developed and confirmed by studies at the Michigan State University[3] was chosen as desirable.

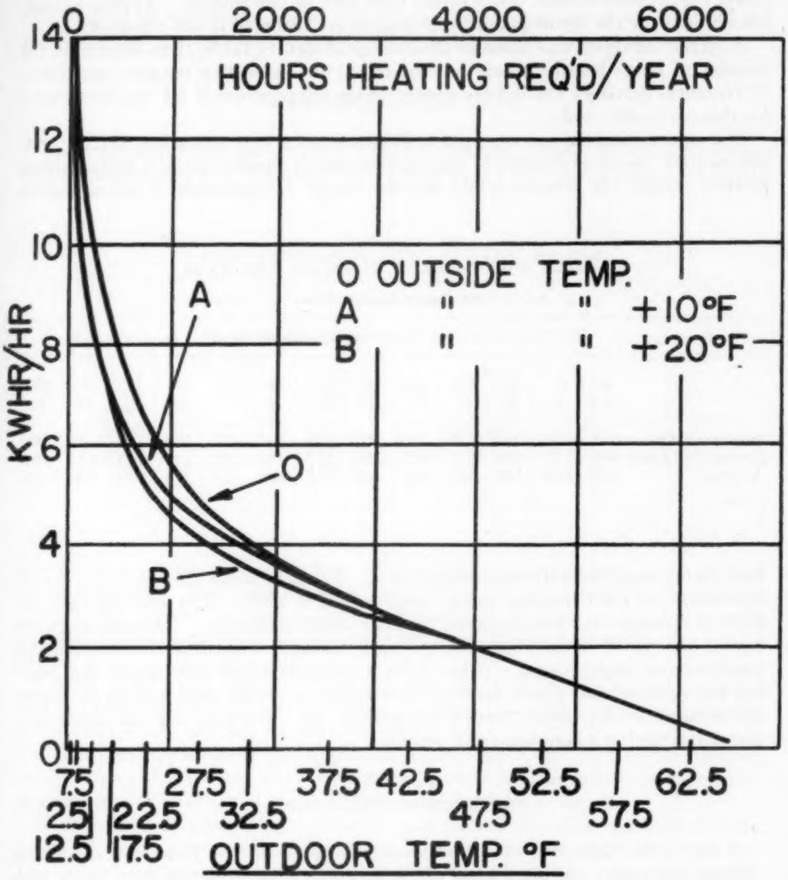


FIG. 5—ELECTRIC ENERGY CONSUMPTION OF AIR-TYPE HEAT PUMP (WITH AND WITHOUT SOLAR SUPPLEMENTATION)

With air temperature increases that could be practically realized, Table 1 was developed. This table indicates temperatures to the heat pump ranging from −7.5 to 62.5 deg F and with 10- and 20-deg average rise through solar collector and/or storage. Using these values for supplemented source conditions, corre-

sponding increases for heat pump CP and capacity were obtained from the manufacturers' performance curves for supplemented source air conditions. A relative decrease in energy consumption of the electrical resistance heaters used as standby is readily apparent. From these factors and the house heating demand, the total power requirement at that temperature level may be determined. Typical sample calculations for the development of this table are included in the Appendix.

A further analysis was made of climatological data[4] (Table 2) to determine the number of hours heating would be required at the respective temperature levels. A Technical Advisory Committee winter design temperature of 5 F was considered for this particular study.

The values obtained and reported in Tables 1 and 2 were plotted as Fig. 5. The curves show the annual hours of heating required at various outdoor temperatures plotted against the corresponding electric energy consumption of an air-source

TABLE 3—NUMBER OF DEFROSTS PER YEAR

(100 Percent Running Time)

	Technical Advisory Committee Temperatures (F)												
	−10 to 5	6-8	9-11	12-14	15-17	18-20	21-23	24-26	27-29	30-32	33-35	36-38	39-40
Normally Dry	475	450	400	375	350	300	250	200	175	150	100	75	50
Normally Damp	800	750	700	625	575	500	425	350	300	225	175	125	100
Average	637	600	550	500	462	400	337	275	237	212	137	100	75

heat pump alone and with solar supplement. The area under the respective curves represents the total seasonal energy requirement in kwhr. The area between the curves represents the energy saved by solar supplementation. The area between curves was found to be 1200 kwhr or approximately 6 percent of the usual unsupplemented requirement. This was for a region in which 6500 degree-day heating was required and where the solar heat added to the air produced an air-source temperature 10 deg above that of the ambient air. For a 20-deg rise above ambient, this saving amounted to 12 percent.

REDUCTION OF COIL DEFROST

A desirable *by-product* of supplementation has been the reduction of defrost periods necessary. Limited observation indicates this reduction may be as high as 75 to 80 percent. It was reported from studies of field installations by manufacturers of equipment that the number of defrost periods is proportional to the TAC design temperature, the generalized normal-humidity level (Table 3)[4] and a balance-point defrost factor (Table 4).[4]

For conventional equipment defrost, heat is extracted from the heated space and rejected to the outdoor coil to melt the frost or ice. This then represents a heat loss to the space. The loss amounts to approximately 2000 Btu per nominal horsepower for each defrost.

When considered for the entire heating season, this amounts to an appreciable heat loss which must be supplied by additional equipment operation.

A reduction approaching 2 percent in operating energy could be realized by eliminating the necessity for defrost. Perhaps more important is the probability of reduction of maintenance and servicing problems occasioned by icing and the possible reduction or elimination of defrosting equipment components.

Results of System Tests

In operations during the winter of 1955-56, without storage, an average increase in performance (CP) of 15 percent and in capacity of 16 percent was noted during several weeks of daytime operation. These results were obtained for a variety of solar-reception conditions ranging from clear to cloudy weather. During the most cloudy period, the increase in performance of the heat pump due to solar supplementation amounted to 6 percent with an increased capacity of 13 to 14 percent.

Table 4—Balance Point Defrost Factor

	Technical Advisory Committee Temperatures (F)		
Balance Point	−10 to 10	11 to 25	26 to 40
10-15	0.50	0.45	0.40
16-20	0.60	0.55	0.45
21-25	0.75	0.65	0.55
26-30	0.85	0.75	0.65
31-35	1.00	1.00	0.80
36-40	1.00	1.00	1.00

Studies during the winter of 1956-57, after the addition of storage, revealed that the solar collector and storage were providing an increase in air temperature to the heat pump varying between 12 and 23 deg.

Continuous operation of the 2 systems during two 3-week periods in January and February showed that the CP of the supplemented heat pump averaged 17.5 percent higher than that of the unit using outside air as a source.

Measurement of total flow (by volume water meter) to a water-cooled condenser combined with temperature rise provided comparable heat-output information. The electric energy input required to produce the effects was measured by calibrated conventional kwhr meters that could be read to 1/100 kwhr accuracy.

Planned Operation

For operations during the winter of 1957-58 the systems are being modified to provide operation in accordance with actual heating demand for existing weather conditions rather than continuous operation as heretofore studied. The blower, under differential-thermostat control, provides circulation of air from the collector

through storage when solar energy is being collected and can be stored. This blower is also interlocked to move air through storage and over the evaporator regardless of whether solar collection is taking place when heating is required.

In theory it should be possible to increase significantly the quantity and temperature level of heat in storage and the temperature rise that can be provided the air en route to the heat pump over that obtained under previous continuous operating conditions.

ECONOMIC CONSIDERATIONS

The results of theoretical and practical studies to date have indicated that the economic advantage of improved equipment operation obtained through the use of solar energy supplementation will be offset by extra construction costs of collectors, storage, and required controls. Depending upon how well these features can be worked into an architectural plan, their costs may be amortized over a 3- to 5-year period.

CONCLUSIONS

Incorporation of a solar collector and storage with the conventional air-type heat pump can be accomplished with a minimum of system addition and change. Use is made of the south-facing building walls as part of collector construction. Yet to be fully assessed, but definitely demonstrated, has been the reduction and simplification of coil defrost problems.

Solar-energy supplementation with storage has provided a heat-source air-temperature increase ranging between 10 and 20 deg for the air-source heat pump. Significant increases in both heat pump capacity and CP have reduced operating costs over those obtained with an identical conventional unit using outside air as a heat source and offer enough advantage to warrant continued development. Further improvements in collector and storage efficiency through design are possible. Whether they may be accomplished without disproportionately increasing supplemental equipment costs relative to operational energy savings is still under study.

REFERENCES

1. ASHVE RESEARCH REPORT No. 1502—Availability and Utilization of Solar Energy, Part I—Soalr Energy Availability for Heating in the United States, by R. C. Jordan and J. L. Threlkeld (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 177).

2. A Review of Solar House Heating, by Maria Telkes (*Heating and Ventilating*, Reference Section, September 1949).

3. Heating Air by Solar Energy, by F. H. Buelow and J. S. Boyd (*Agricultural Engineering*, January 1957).

4. Chart 2, Bulletin AM 77-2954 2m (86), General Electric Co.

APPENDIX

Heating requirements for a house of approximately 12,000 cu ft with a heat loss as indicated in Fig. 4 were anticipated.

Sample calculations are for the kwhr per hour required for an outdoor temperature, t, where $5F < t < 10$ F.† From Table 1 and Fig. 4 and 7.5 F the heat pump capacity alone is 31.3 M Btu per hr at a CP of 1.72. The energy for operation of the heat pump is therefore 31,300/ 3,413(1.72) = 5.33 kwhr.

Referring again to Fig. 4, it was apparent at this temperature that the house assumed would require 53 M Btu per hr. The difference between 53 and 31.3 M Btu per hr must be supplied by electrical resistance or 53 − 31.3 = 21.7 M Btu per hr.

Therefore, the energy required to produce this heating effect is 21,700/3,413 or 6.35 kwhr per hr. When combined with the energy for operation of the heat pump, a total energy of 5.33 + 6.35 or 11.68 kwhr per hr was determined.

Similarly, where a 10-deg temperature rise results from solar collection, the capacity of the heat pump now corresponds to that at which the outdoor temperature is 15 F $< t <$ 20 F in which case the capacity of the unit is 38 M Btu and the CP 1.92. For heat pump operation, as calculated previously, 5.8 kwhr per hr electrical energy is required. The heat loss of the house remains at 53 M Btu per hr of which 53 − 38 = 15 M Btu per hr must now be supplied by electrical resistance heating or 4.4 kwhr per hr.

A new factor of 0.28 kwhr per hr was calculated as additional fan energy necessary to move the air through the collector and storage to the heat pump unit. The total energy consumed, therefore, for the 10-deg rise is 5.8 + 4.4 + 0.28 = 10.48 kwhr per hr.

DISCUSSION

F. R. O'BRIEN*, Birmingham, Ala. (WRITTEN): It seems probable that the first utilization of solar energy on any large scale will be in conjunction with the heat pump for space heating. It thus seems important to devote some real research effort to this form of utilization of solar energy in order to develop the controlling parameters.

We have had in the past few years the excellent studies relating to the availability and utilization of solar energy, which have been published in the ASHAE TRANSACTIONS. These studies have certainly aroused interest in the solar-heat—heat-pump possibility, enough to result in several applications of this principle and in the research installation at Kansas State College.

The full-scale applications are of interest, because they furnish proof of the possibility, but it must be admitted that the information to be gained will be limited and perhaps difficult to translate into design data.

The research work discussed in this paper will ultimately be of considerable value, because the experimental apparatus has been designed in such a manner as to permit flexibility of operation with a consequent ability to explore various parameters.

It is interesting to note that the authors arrive at the same conclusions as Jordan and Threlkeld that low-side storage is desirable and that a storage medium with a melting point in the neighborhood of 50 F is desirable. This should emphasize the need for research leading to the development of an inexpensive material with the proper fusion temperature and a heat of fusion between 150 to 200 Btu. I had some difficulty with the values reported in Table 1 and would suggest that this table be reconsidered with a view to its effect on the conclusions drawn from Fig. 5.

A generalization can perhaps be made that the state of solar-energy utilization in conjunction with heat pump is at about the same point where heat pumps were in the thirties. With an increased research effort, perhaps the state of the art can be ad-

†Mathematical symbolism that t lies between 5 F and 10 F or, stated another way, that t is greater than 5 F and less than 10 F.

*Assistant Director, Southern Research Institute.

vanced to keep pace with heat pumps themselves. The authors should be commended for their part in this advancement.

AUTHORS' CLOSURE (Mr. Davis): Discussion with Mr. O'Brien subsequently further interpreted and explained changes in heat-pump capacity of Table 1 for outside air temperatures of 12.5 F through 22.5 F. These were due to the particular performance characteristics of the commercial equipment analyzed and were concerned with shift from three- to two-cylinder compressor operation as indicated in Fig. 4. It was mutually agreed the hypothetical calculations are correctly made, though analysis of other manufacturers' equipment would provide slightly different performance characteristics at these temperature levels.

1626

No. 1626

COOLING LOAD FROM THERMAL NETWORK SOLUTIONS

by HARRY BUCHBERG*, LOS ANGELES, CALIF.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Department of Engineering, University of California, Los Angeles.

THE STUDIES described in this paper are an extension of work reported on previously[1, 2]. In these studies an evaluation was made of the importance of many influencing factors on the sensible cooling load in a light frame test house subjected to the time variable climate. Among the factors considered were:

1. The separate solar inputs for the various wall orientations and the roof.
2. The diffuse solar power transmitted by shaded glass areas.
3. Long-wave radiation exchange with the surroundings.
4. Radiation exchange between inside surfaces.
5. Heat sources within the space.

METHODS USED

The factors listed were evaluated by obtaining electric analogue solutions of the thermal network representing the test house heat transfer system. It will be noted that all of the factors mentioned are concerned with the boundary conditions imposed on the network and with the manner in which the heat conduction paths through the structure are coupled on the inside. In addition, several different conduction path representations were investigated. These comprised a fine net and various degrees of simplification.

SYSTEM INVESTIGATED

The system investigated was the one-room wood frame dwelling described in detail in Reference 1 and shown schematically in Fig. 1. To conserve space, the details of construction will not be repeated here. Likewise, the thermal network principles and idealizations made in drawing the network, shown in Fig. 2, were

---

* Associate Professor of Engineering, Department of Engineering, University of California.
[1] Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, January 1958.

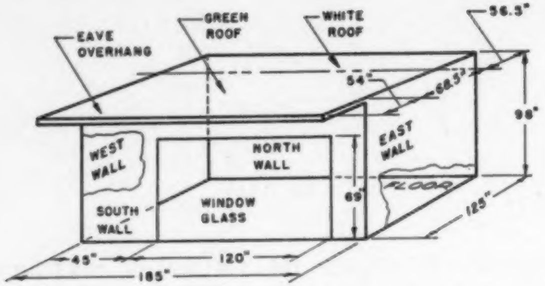


FIG. 1—SCHEMATIC DIAGRAM OF TEST HOUSE

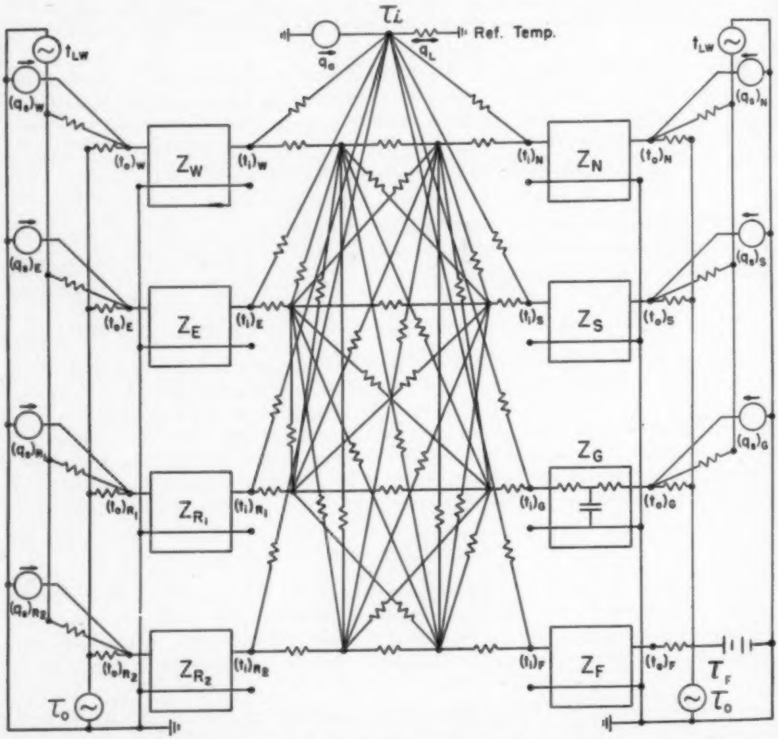


FIG. 2—BASIC THERMAL NETWORK REPRESENTING THE IDEALIZED TEST HOUSE

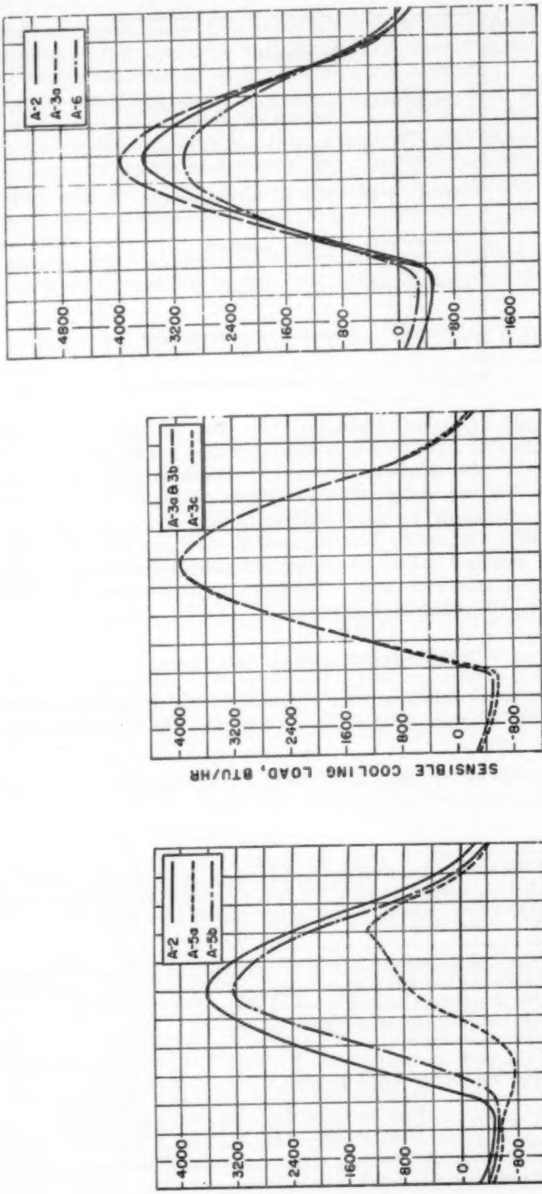
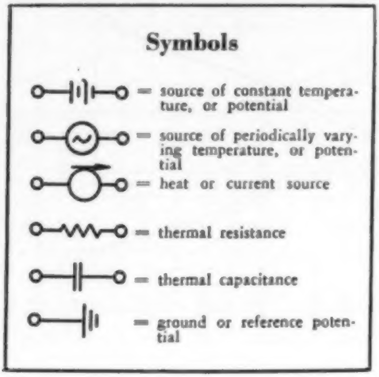


FIG. 3—COOLING LOAD CURVES BASED ON SOLUTION OF THERMAL NETWORK CONFIGURATION A



presented in detail in earlier publications[1, 3]. The thermal network shown in Fig. 2 differs from that given in Fig. 3 of Reference 1 in two respects. *First*, the conduction paths are shown here as 4-terminal networks labeled Z_W, Z_E, etc., without showing the R-C net, except for Z_G. Table 1 presents a description of the conduction path impedance networks used in various solutions. *Second*, for purposes of determining the cooling load required to maintain the space temperature constant at the specified value, the inside air temperature point, τ_i, is shown grounded through a small resistance. The input potentials were then all reckoned with respect to τ_i, the reference temperature.

Thermal Network Solutions

The thermal network solutions were obtained with a dc electric network computer known as the Thermal Analyzer[4]. As shown in Fig. 2, the boundary inputs

Table 1—Description of the Conduction Path Network Configurations[a]

Configuration Designation	Description
A	The conduction path networks are the same as those shown in Fig. 3, Ref. 1, consisting of a *fine* net of parallel tee branches representing the paths through wall studs or ceiling joists and the space between structural members.
B	Simplified conduction path networks in which the parallel paths are replaced by a single path, considering the frontal area taken by the structural members as additional air space, but maintaining the same fineness of lumping as configuration A.
C	The same as configuration B except that the number of lumps representing each conduction path was reduced as follows: Wall Wections: Reduced from 6 to 3 lumps Roof Sections: Reduced from 3 to 2 lumps Floor Section: Reduced from 2 to 1 lump

[a] For the network diagrams, see Figs. A-1, 2, and 3 in the Appendix.

comprised the diurnal ambient air temperature τ_0, a time variable longwave radiation temperature t_{LW}, and time variable thermal current inputs q_s, representing the solar energy absorbed by each of the exposed surfaces.

The special equipment used to provide the necessary time-variable inputs and the actual values used are given in Reference 1. The only additional equipment was a dividing resistance network and constant potential source used to provide the input currents representing the diffuse solar power transmitted through the shaded glass area and absorbed on all surfaces *seen* by the sky. In some cases all of the current was directed either to the inside surface of the north wall or the floor as noted.

The network solutions consisted in measuring the potential τ_i above ground (proportional to the sensible cooling load) and other potentials in the network

TABLE 2—DESCRIPTION OF ELECTRIC ANALOGUE SOLUTIONS

SOLUTION DESIGNATION	DESCRIPTION
— 1a	Prediction of τ_i, dependent variable $(q_{sd})_G$, diffuse solar radiation transmitted by glass, neglected
— 1b	Prediction of τ_i, dependent variable $(q_{sd})_G$, included in network.
— 2	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$, neglected
— 3a	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ included in network
— 3b	Prediction of sensible cooling load, τ_i = 76 F Total $(q_{sd})_G$ directed into floor
— 3c	Prediction of sensible cooling load, τ_i = 76 F Total $(q_{sd})_G$ directed into north wall
— 5a	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ and (q_s) for all exposed surfaces neglected
— 5b	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ neglected, $(q_s)_{R1 \text{ and } R2}$ (Solar radiation absorbed by roof) included but all other solar inputs neglected
— 6	Prediction of sensible cooling load, τ = 76 F $(q_{sd})_G$ and inside radiation exchange network neglected
— 7	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ and q_s (inside heat source) neglected
— 9	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ and outside longwave radiation exchange neglected
— 10a	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ and $(q_s)_w$ (west wall solar input) neglected
— 10b	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ and $(q_s)_{R1 \text{ and } R2}$ (roof solar input) neglected
— 10c	Prediction of sensible cooling load, τ_i = 76 F $(q_{sd})_G$ and $(q_s)_E$ (east wall solar input) neglected

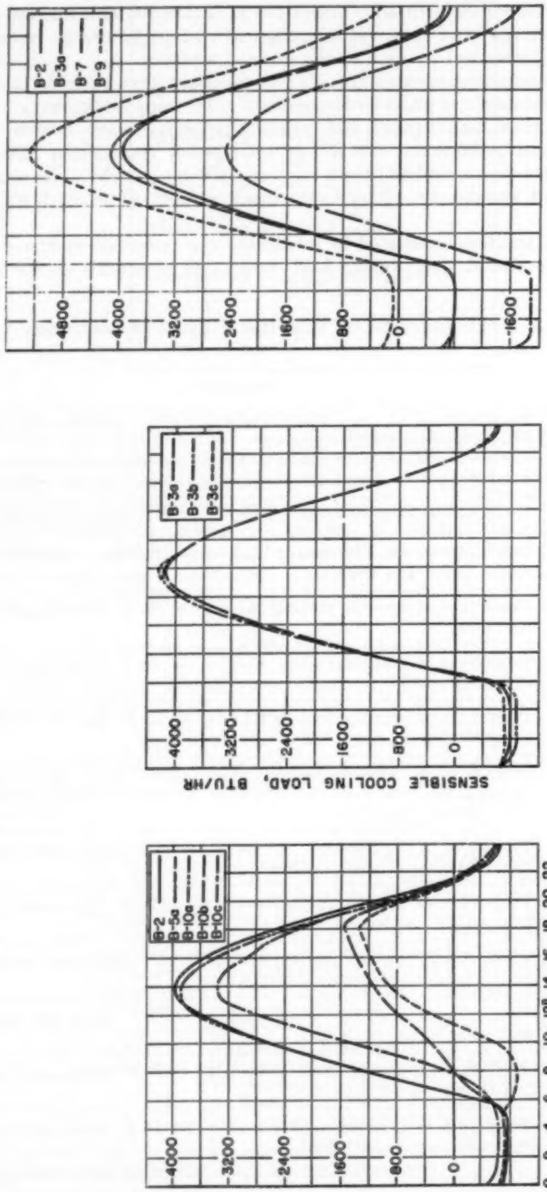


Fig. 4—Cooling Load Curves Based on Solution of Thermal Network Configuration B

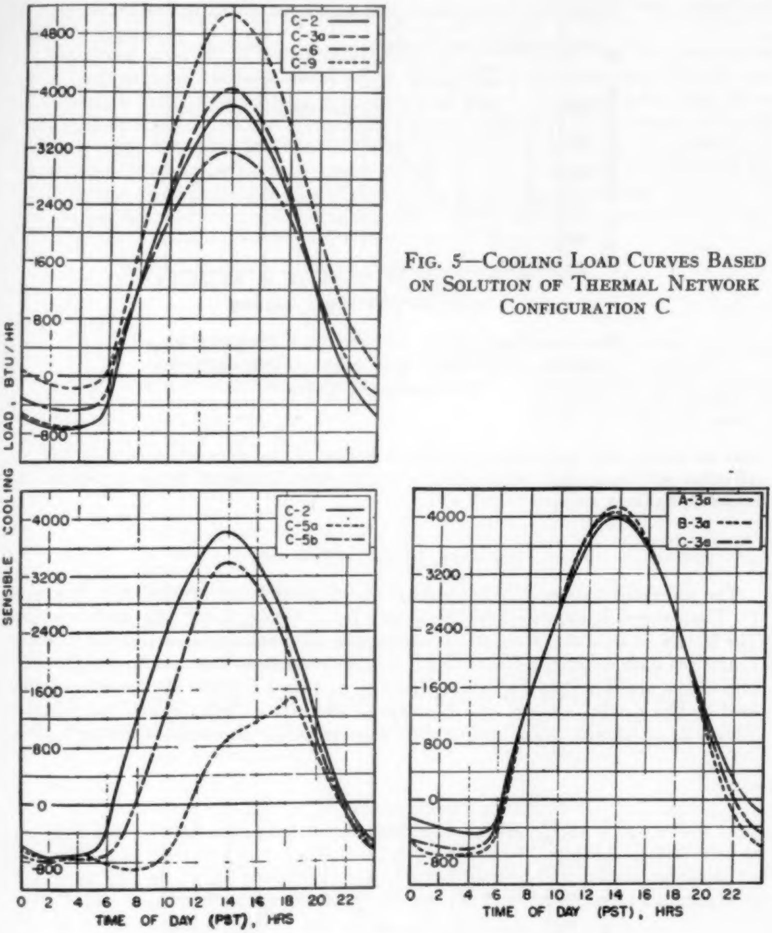


Fig. 5—Cooling Load Curves Based on Solution of Thermal Network Configuration C

corresponding to surface temperatures. The resistance through which the inside air potential point was grounded was selected to give a maximum variation in τ_i that was at least an order of magnitude smaller than the reference or inside air temperature (76 F or 76 volts). The scale factors between the thermal and electrical analogous quantities were the same as those reported in Table 1 of Reference 1 and are given in Table A-1 in the Appendix. A list of the solutions obtained is given in Table 2. Solutions-1a and-1b of the network were obtained considering τ_i as a dependent variable. For this case the τ_i point was grounded through a capacitor representing the thermal capacitance of the air in the room. It should

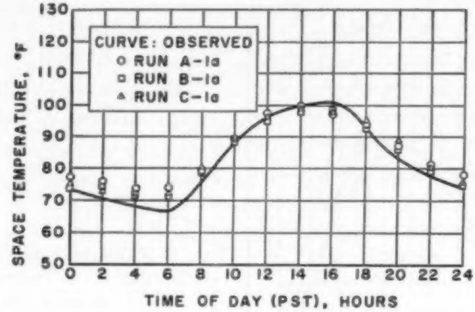


FIG. 6—PREDICTION OF SPACE TEMPERATURE-VARIOUS NETWORK SOLUTIONS COMPARED TO EXPERIMENTAL DATA

also be noted that solution —2 is the solution of the network shown in Fig. 2. All other solutions after —2 required some modification in the boundary inputs as noted in Table 2.

COMPUTER RESULTS DISCUSSED

The solutions described in Table 2 are shown graphically in Figs. 3, 4, 5, 6, and 7. Each curve is identified in the legend by a capital letter and dash number. The letters A, B, and C refer to the conduction path configurations, given in Table 1, and the dash number refers to the boundary conditions imposed during the solution of the network (see Table 2). In Figs. 3, 4, and 5 where the sensible cooling load in Btu per hr is given as a function of time for a 24-hr period, the separate influences of various solar inputs, diffuse solar power transmitted through glass,

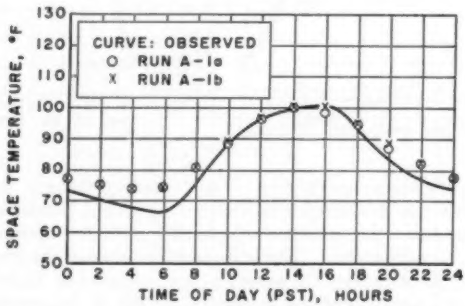


FIG. 7—PREDICTION OF SPACE TEMPERATURE ILLUSTRATING THE INFLUENCE OF DIFFUSE SOLAR RADIATION TRANSMITTED BY GLASS AREA

longwave radiation exchange inside and outside, and the inside heat source are readily apparent.

Beginning with Fig. 3 configuration A, (the fine net) one may compare curves A-2, A-3a, and 6. Using solution —2 as the standard it can be seen that the predicted peak instantaneous load is increased by about 9 percent when the diffuse solar power transmitted through the glass is accounted for. The increase in daily cooling load, however, is about 5 percent. Solutions with the simplified configurations B and C indicate the same trend but the differences in peak load are slightly less than predicted with the fine net. Fig. 7 indicates that inclusion of diffuse transmitted solar power had the effect of increasing the prediction of space

NOMENCLATURE

C = heat capacity, Btu per Fahrenheit degree.
E = electrical potential, volts.
i = electrical current, amperes.
q = heat flux, Btu per hour.
q_h = interior heat source, Btu per hour.
q_L = sensible cooling load, Btu per hour.
q_s = solar input or direct solar energy absorbed by an exposed surface. Btu per hour.
$(q_{sd})G$ = diffuse solar radiation transmitted by the glass area, Btu per hour.
R = resistance to heat transfer, Fahrenheit degrees per Btu per hour.
t_i = inside surface temperature, Fahrenheit.
t_0 = outside surface temperature, Fahrenheit.
t_{LW} = outside long-wave radiation potential, Fahrenheit.
Z = conduction path thermal impedance, Fahrenheit degrees per Btu per hour.

Greek Letters

θ = time.
τ_i = inside ambient air temperature, Fahrenheit.
τ_0 = outside ambient air temperature, Fahrenheit.

Subscripts

E = east wall.
F = floor.
G = glass.
N = north wall.
R_1 = white roof section.
R_2 = green roof section.
S = south wall.
W = west wall.

temperature by 1 to 2 deg for most of the day. From Figs. 3 and 4 it is quite apparent that the transmitted diffuse solar power can be handled equally well by assuming that all of the power is incident on either the floor or wall opposite the glass. The load prediction, particularly with the fine net, was indistinguishable for the 3 solutions, —3a, —3b, and —3c.

The separate influences of the various solar inputs may now be examined. A comparison of solutions —5a, —5b, —5c, —10a, —10b, and —10c with solution —2 in Figs. 3, 4, and 5 illustrates dramatically the importance that solar inputs to the various exposed surfaces can have on cooling load predictions. The total daily cooling load, without considering the solar inputs through all opaque walls, was only about 22 percent of the total daily load including all inputs. Other interesting observations that can be made regarding the effect of solar inputs are:

1. For exposed roofs, solution —5b and —10b indicate clearly that the roof solar input is the most important of all and must never be neglected, in this instance accounting for almost 60 percent of the total daily load.

2. Solution 10a indicates the value of wall insulation in the reduction of cooling loads. The *insulated* west wall accounts for only a very small part of the total daily load while the uninsulated east wall accounts for about 16 percent.

3. It should also be noted that the solar input to the east wall is much more effective in increasing the peak instantaneous load, building up during the morning hours, while the west wall solar load merely widens the load curve increasing the load in the afternoon and evening and night depending on the time constant of the wall. For buildings having primarily day occupancy, attention to the east exposure is important in cooling load considerations.

4. The effect of south wall solar input does not appear in this investigation because the south exposure was shaded by eaves overhang.

TABLE 3—TOTAL DAILY COOLING LOAD AND NET 24-HR HEAT TRANSFER

CURVE NO.	DAILY COOLING LOAD, BTU	24-HR NET HEAT TRANSFER, BTU
A-2	34,900	32,100
B-2	37,500	32,170
C-2	36,800	32,140
A-3a	36,700	33,890
B-3a	39,500	33,610
C-3a	38,400	33,730

The influence of longwave radiation exchange on cooling load is considerable. Comparing load curve B-9 with B-2 in Fig. 4 it can be seen that the predicted peak instantaneous load is increased by about 34 percent and the total daily load by about 53 percent when longwave radiation exchange with the surroundings is neglected. The effect of neglecting longwave radiation exchange between the interior surfaces can be seen by comparing load curve A-6 with A-2 in Fig. 3. The effect was to reduce the predicted peak instantaneous load by about 15 percent and the total daily load by about 9 percent. As the interior surface temperature differences become less the importance of considering radiation exchange becomes less.

As would be expected constant space heat sources have the effect of changing the dc level of the load curve. Comparing load curve B-7 with B-2 in Fig. 4 indicates clearly the dc shift down when neglecting the constant inside source, q_a.

A comparison is made of solutions —3a for configurations A, B, and C in Fig. 5. Solution —3a includes all inputs. The maximum deviation in the peak instantaneous load between any of these solutions is only 4 percent; for the —2 solutions it is about 8.5 percent. Table 3 presents a comparison of the total daily cooling loads and net heat transferred during the 24-hr period or the net area under the load curve. As would be expected, the net heat transferred during a 24-hr period is substantially the same for all configurations for a specified set of boundary inputs. However, there are some differences in the total daily cooling load predicted. The maximum deviation between solutions is about 7 percent.

The space temperature predictions with configurations A, B, and C are compared in Fig. 6. The simplified configuration actually appears to give a slightly better prediction for the night hours than the fine net, A. However, this result is not significant. It can be stated that the differences in prediction between the various

configurations appear to be less than the overall errors possible in this kind of calculation.

Conclusions

1. These studies demonstrate once more the flexibility in computing with a dc electric network computer when the influence of many variables in a complex thermal system is being sought.

2. It is apparent from these studies that fairly course networks can be used to achieve reasonable engineering accuracy in the prediction of cooling loads. A more detailed study of network *lumping* and its influence on the magnitude and phase angle of the impedance function for several typical wall constructions is now in process.

3. The effect of diffuse solar power transmitted through shaded glass areas on the cooling load prediction can adequately be included in a network solution by simply assuming that all of the power is incident on either the floor or the wall opposite the glass area, whichever is more convenient.

4. Solar inputs to opaque walls and roof can be very important in the prediction of cooling loads particularly for small uninsulated structures with several exposures and windows shaded from direct sunlight. All solar inputs should be evaluated before any are deleted from the network.

5. Longwave radiation exchange with the surroundings can influence greatly the predicted cooling load, particularly when exterior building surface temperatures rise appreciably due to incident solar power.

6. Longwave radiation exchange between interior surfaces usually does not have a large influence on cooling loads, but may have a significant effect and should be evaluated before being deleted in any calculation. Inside radiation exchange will have a greater effect when significant radiation sources, such as luminaires are present in a room. Radiation sources within an enclosure may be treated as thermal current inputs to the inside surfaces exposed. This may be done by means of a dividing network in a manner similar to that used for diffuse solar radiation transmitted by glass areas.

Acknowledgment

Undergraduate students assisted in the machine calculations and reduction of data.

References

1. ASHAE Research Report No. 1543—Electric Analogue Prediction of the Thermal Behavior of an Inhabitable Enclosure, by Harry Buchberg (ASHAE Transactions, Vol. 61, 1955, p. 339).

2. ASHAE Research Report No. 1565—Electric Analogue Studies of Single Walls, by Harry Buchberg (ASHAE Transactions, Vol. 62, 1956, p. 177).

3. ASHVE Research Report No. 1497—Circuit Analysis Applied to Load Estimating, by H. B. Nottage and G. V. Parmelee (ASHVE Transactions, Vol. 60, 1954, p. 59).

4. The Thermal Analyzer, A Special Purpose Computer, by W. L. Martin and R. Bromberg (*Proceedings of the Electronic Computer Symposium*, University of California, Los Angeles, April 30, 1952).

APPENDIX

A summary of scale factors used in the electric analogue solutions of the thermal networks is given in Table A-1. The thermal network configurations representing the

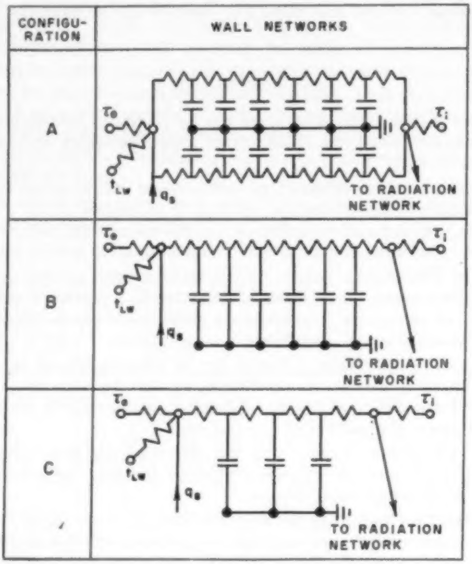


FIG. A-1—CONDUCTION PATH THERMAL NETWORK CONFIGURATIONS FOR WALL SECTIONS

TABLE A-1—SUMMARY OF RATIOS AND UNITS OF ANALOGOUS ELECTRICAL AND THERMAL QUALITIES

QUANTITY	UNITS		SCALE FACTORS	
	THERMAL	ELECTRICAL	RATIO	VALUE
Time	hrs	sec		2
Capacity	$\frac{\text{Btu}}{\text{F deg}}$	Farads	$\frac{C_t}{C_e}$	8 x 10^6
Resistance	$\frac{\text{F deg}}{\text{Btu/hr}}$	Ohms	$\frac{R_e}{R_t}$	16 x 10^6
Potential	F deg	Volts	$\frac{E}{t - t^a}$	1
Rate of Energy Transfer	$\frac{\text{Btu}}{\text{hr}}$	$\frac{\text{Coulombs}}{\text{sec}}$ or Amperes	$\frac{q}{i}$	16 x 10^6

t^a = Reference temperature. Subscripts: t = thermal circuit element, e = electrical circuit element.

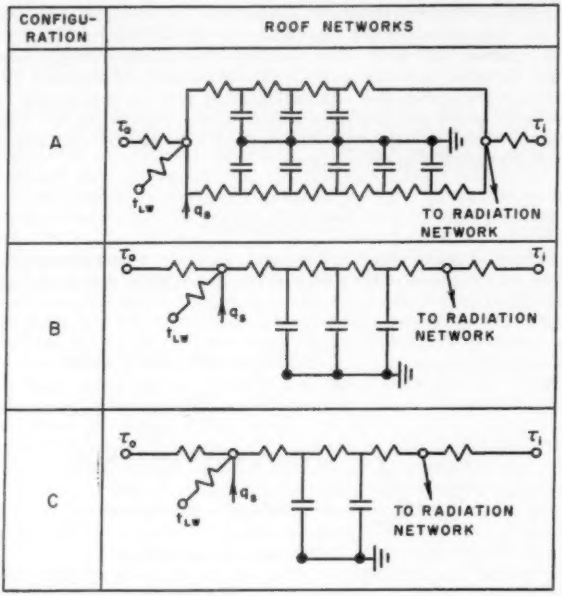


FIG. A-2—CONDUCTION PATH THERMAL NETWORK CONFIGURATIONS FOR ROOF SECTIONS

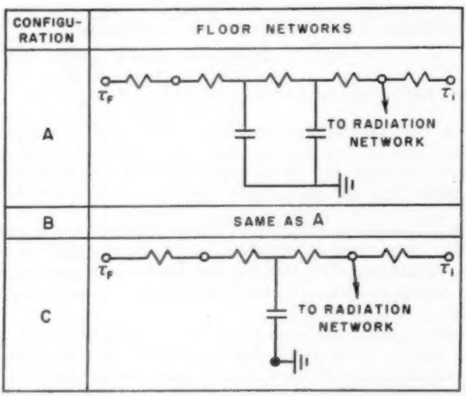


FIG. A-3—CONDUCTION PATH THERMAL NETWORK CONFIGURATIONS FOR FLOOR SECTION

conduction paths of separate structural components are shown in Figs. A-1, A-2, and A-3. Each of these networks was represented by an impedance (Z) in the complete thermal circuit shown in Fig. 2. The dividing thermal network representing diffuse solar radiation absorbed and transmitted by the glass area and absorbed by the interior wall surfaces is shown in Fig. A-4. It will be noted that the diffuse solar radiation absorbed by the various surfaces was treated as a thermal current input. The resistances in the dividing network depend on the shape factors between the surfaces and sky and on the radiation properties of the surfaces.

DISCUSSION

H. B. NOTTAGE, Encino, Calif., (WRITTEN): In order to achieve practical applications of the simplifying procedures shown to be reasonable in this study, and under the

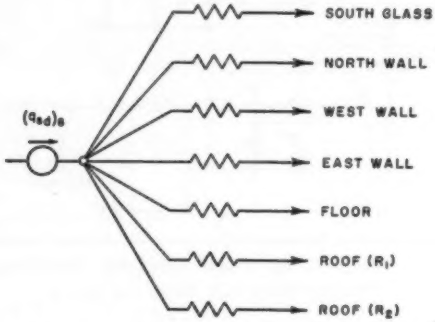


FIG. A-4—THERMAL DIVIDING NETWORK REPRESENTING DIFFUSE SOLAR RADIATION ABSORBED AND TRANSMITTED BY GLASS AREA AND ABSORBED BY INTERIOR WALL SURFACES

usual circumstance that an analogue would not be available (and perhaps uneconomic if one should be available), the practicing engineer would appreciate guidance by having the author present a comparison of the analogue solutions with predictions made by THE GUIDE method. Can this be added?

L. W. NELSON, Minneapolis Minn., (WRITTEN): Because the solar heat gain on the roof is such a large percentage of the total heat gain, it would be very interesting to know the effect of reducing this gain by the various recommended procedures, *e.g.*, attic ventilation, ceiling insulation and a reflecting surface on the underside of the roof in an attic space.

I would like to know what the diurnal outdoor temperature variation was during the predicted cooling loads. If the same conditions were used as in Reference 1 the heat flux through the inside surface of the east wall would be considerably less, thus showing a less important influence on the insulating properties of the east wall. Also in Reference 1 the west wall is shown to have a greater heat flux than the east wall during the peak load condition even though it is insulated. This would seem to show the great

need for insulating the west wall particularly in light residential frame construction where the time lag is relatively small.

I wonder if the simplification of the thermal network is the reason for this difference or if it can be attributed to something else.

G. V. PARMELEE, Dhahran, Saudi Arabia, (WRITTEN): Mr. Buchberg's detailed study of a thermal network is the sort of research that increases understanding of how thermal systems react to external influences. I think that it is important to realize that he has worked with a linear system, that is, one in which resistances and capacitances are not influenced by temperature differences or temperature levels. Hence, he has been able to treat input functions individually and to show their significance in the total load. It is this individual treatment of input functions that will have to be followed in the development of application for load estimating, a matter which was stressed by Gilman[A] and Clausen in their recent paper.

Every circuit solution that is presented is one more step toward building up a body of estimating data, and these solutions are extremely useful in that respect. However, if one is tempted to try to apply these results to practical problems, one should note that (1), they are for a very light structure and (2), the constant indoor heat source of about 1800 Btuh accounted for about 45 percent of the peak load. The first point affects time lags and damping effects. The second affects ratios of heat flow rates, but it does not affect the shapes of the curves.

In connection with developing application data from circuit solutions, it seems appropriate to describe a procedure which I worked out about a year ago. This was a method of estimating cooling loads from instantaneous rates of heat gain, such as the tabular data given in THE GUIDE. The attempt was crude, but the basic principle, with refinements, might be found adequate for estimating purposes. The method was to use a *decrement factor* for the damping effect of the heat capacity of the interior of a building on the instantaneous rate of heat flow, together with a time lag. The decrement factor was developed as follows.

Stewart[B] has shown that the instantaneous rate of heat flow through a wall or a roof can be estimated with reasonable accuracy by the following equation:

$$q = UA(t_m - t_i) + \lambda_1 UA(t_e - t_m), \text{ Btuh} \quad . \; . \; . \; . \; . \; . \; . \quad \text{(A)}$$

where

U = the overall coefficient of heat transfer, Btuh (sq ft) (deg F)
A = area, sq ft
t_m = 24-hour average sol-air temperature, F deg
t_i = the constant indoor temperature, F deg
t_e = the sol-air temperature, F deg, at an hour earlier than the hour in question by the time lag of the wall
λ_1 = a decrement factor, dimensionless

The decrement factor, λ_1, is made up of parts of the fundamental and second harmonic decrement factors as found in the charts prepared by Mackey and Wright[C]. The quantity $(t_e - t_m)$ is very roughly the fundamental harmonic of the sol-air temperature wave. The time lag is the sum of the fundamental time lags for each material in the wall or roof.

It follows, then, that it should be possible to express cooling loads for both walls and glass by introducing an additional decrement factor into each term to account for the additional damping of the interior structure. An additional time lag would be included in selecting t_e in the second or periodic component of the equation. Thus, cooling loads would be found as follows:

for walls and roofs:

$$q = \lambda_2 UA(t_m - t_i) + \lambda_1\lambda_2 UA(t_e' - t_m), \text{ Btuh} \quad . \; . \; . \; . \; . \; . \quad \text{(B)}$$

for glass transmitting solar radiation:

$$q = \lambda_3 A(I_m \tau_m) + \lambda_2 A(I_e' \tau_e' - I_m \tau_m), \text{ Btuh} \quad \text{(C)}$$

where

t_e' = sol-air temperature, F deg, at an hour earlier than the hour in question by the time lag of the wall or roof plus the additional time lag due to the heat capacity of the interior structure
λ_2, λ_3 = decrement factors due to the damping effect of the interior structure
$I_m \tau_m$ = 24-hr average transmitted solar radiation, Btuh
I_e, τ_e' = solar radiation transmitted at an hour earlier than the hour in question by the time lag due to the heat capacity of the interior structure, Btuh

The obvious method for developing the additional decrement factors and time lags was to make comparisons between the results of solving circuit problems and calculated

TABLE A—MAGNITUDE OF DECREMENT FACTORS

	λ_2	λ_3	ADD. TIME LAG
A. Solar radiation transmitted through north glass ref. D	0.54	0.95	3 hrs
B. Solar radiation transmitted through south glass ref. E 2 in. floor	0.66	1.00	1 hr, approx.
6 in. floor	0.47	1.00	1 hr, approx.
C. Southwest wall ref. F	0.93	0.92	0 hr
D. Roof problem	0.34	0.83	3 hr

rates of instantaneous heat flow. It is hardly necessary to say that there were few circuit results to use. Furthermore, only the loads which resulted from single input functions could be used. The circuit cooling load was re-expressed as a 24-hr average plus a periodic component. Ratios of periodic load component to periodic instantaneous heat gain component were then determined, with due allowance for additional time lag. Results from 4 solutions which were available gave a rough idea of the magnitude of the decrement factors and are summarized in Table A.

Case D was obtained from a simplified case of heat flow through a roof to an enclosure. For the instantaneous heat flow, the enclosure walls and floor were at the same temperature as the room air. There was no heat flow through them and they had no mass. In computing the cooling load, the walls and floor were assigned mass. A simple circuit was set up with one capacity lump for the roof in both cases and with one additional capacity lump for the floor and walls in the second case. The input function was a sine wave. The material properties are not important to this illustration. The solution was made by A. N. Cerny of the ASHAE Laboratory by means of differential equations.

The decrement factors, λ_2, in table A are 24-hr averages. Hourly values varied considerably, but mainly at the two points of reversal in the sign of the periodic component, where there were usually anomalous negative values. The surprising thing is that the average value of λ_2 gave the cooling loads with 10 percent error or less for a period of 6 to 8 hours long, centered at the time of the peak load. As you might expect, the errors were large during the night hours for Cases A, B, and C. The nature of Case D is

such that the load curve is a sine wave and λ_2 is a true constant. Added time lags were obtained from inspection of the curves. In Case B, λ_2 is unity because of the way the problem was defined. In Case C, λ_2 is high probably because of the panel-cooled ceiling and the generally light construction.

It would be of interest to apply this method to other data, and to refine it by considering harmonics. This and similar analyses would be facilitated if future solutions of circuit problems were expressed in terms of individual inputs and their corresponding loads.

REFERENCES

A. Thermal Circuit Analysis for Developing Application Engineering Information, by S. F. Gilman and O. W. Clausen (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 313).

B. Solar Heat Gain Through Walls and Roofs for Cooling Load Calculations, by J. P. Stewart (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 361).

C. ASHVE RESEARCH REPORT No. 1255—Periodic Heat Flow-Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 50, 1944, p. 293).

D. ASHAE RESEARCH REPORT No. 1529—Circuit Analysis Applied to Load Estimating, Part II—Influence of Transmitted Solar Radiation, by H. B. Nottage and G. V. Parmelee (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 125).

E. Cooling Load From Sunlit Glass, by C. O. Mackey and N. R. Gay (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 321).

F. ASHAE RESEARCH REPORT No. 1595—Analysis of an Air-Conditioning Thermal Circuit by an Electronic Differential Analyzer, by G. V. Parmelee, P. Vance and A. N. Cerny (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 129).

AUTHORS' CLOSURE: Mr. Nelson raises some interesting questions regarding the effectiveness of various devices to reduce roof solar heat gains. To determine the effectiveness of attic ventilation, ceiling insulation, and reflecting surfaces on the underside of a roof, it would be necessary to add to the total network the thermal network for the particular type of roof structure being considered, taking these measures into account. Solutions of the network could then be obtained for various ventilation rates and amounts of insulation. In our experience the most effective and economic method of reducing roof solar heat gains is the use of white exterior surfaces.

In regard to the influence of solar heat gains through the east and west walls for light frame construction, it is clear that the solar heat gain through the uninsulated east wall significantly increased the load during the morning hours as well as the peak load. This is consistent with Fig. 11 of Reference 1 where the heat flux through the east wall is seen to rise rapidly beginning at 6:00 a.m. reaching a peak at about 10:00 a.m. After 12:00 noon, the sun is no longer incident on the east wall but is incident on the west wall which caused the west wall heat flux to increase only slightly above the east due to the insulation. Without insulation a large increase in heat flux (similar to the east wall in the morning) would have occurred. As mentioned in the second paragraph under *Thermal Network Solutions*, all time-variable inputs were the same as given in Reference 1. As Mr. Nelson points out, west wall insulation is valuable for light frame constrution to reduce the afternoon and evening cooling load during the summer months. The value of east wall insulation in reducing the peak load should not be overlooked.

Mr. Nottage would like to have a comparison made between the analogue solution and predictions based on existing GUIDE methods. The author agrees that this would perhaps be desirable but is unable to find the time to make such calculations. This might be a good coordinating job for personnel at the ASHAE Research Laboratory to undertake.

It was certainly of interest to hear from Mr. Parmelee who has for many years been an active supporter of the thermal network approach to heat transfer problems in air conditioning. Mr. Parmelee points out the importance of determining the separate

influence of weather factors on the resulting loads and the author concurs with him. The constant indoor heat source mentioned by Mr. Parmelee was 1280 Btu per (hr) rather than 1800 (See Reference 1, pg. 367). Mr. Parmelee has presented an interesting approach to the development of application data which is really another subject. Three subsequent papers that have a bearing on this subject have been submitted. Comments will therefore be withheld at this time.

1627

No. 1627

THE ASHAE AIR-BORNE DUST SURVEY

By K. T. Whitby*, A. B. Algren**, R. C. Jordan† and J. C. Annis‡, Minneapolis, Minn.

This paper is the result of research sponsored by the American Society of Heating and Air-Conditioning Engineers, in cooperation with the Mechanical Engineering Department, University of Minnesota, Minneapolis, Minn., and the U.S. Public Health Service.

Within the last decade numerous air-borne dust surveys have been made in connection with industrial health and air pollution studies. A study of these surveys shows that they have yielded only limited data on the physical properties of air-borne dusts that have bearing on their removal by air cleaners.

Thus, when the cooperative research project on rating and testing methods for air cleaners for occupied spaces was begun, a search of the existing literature revealed that inadequate data existed on which to base the selection of suitable test dusts for simulated air cleaner testing. A limited dust survey, therefore, was organized, having the following objectives:

1. Determination of the concentration of air-borne dust in typical indoor and outdoor locations by weight and dust spot. An independent measure of the fibrous air-borne particles such as lint was also included, because these largely determine the dust holding capacity of many types of filters.

2. Determination of the volume-size distribution of air-borne dust. The volume distribution was desired rather than the number distribution obtained by microscope count, because filter life is determined by the volume concentration.

3. Determination of other physical characteristics which it was thought might have a bearing on air cleaner performance. Some of these are: density, particle shape, porosity of a packed bed of dust and staining power.

4. Measurement of dust fall at the sampling site. This was desired because the dust fall on indoor surfaces is sometimes the principal nuisance experienced by the housewife.

---

*Assistant Professor, Mechanical Engineering Department, University of Minnesota.
**Professor, Mechanical Engineering Department, University of Minnesota. Member of ASHAE.
†Professor and Head, Mechanical Engineering Department, University of Minnesota. Member of ASHAE.
‡University of Minnesota.
Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, January 1958.

A study of these objectives within the framework of available funds, facilities and personnel led to development of several new dust sampling and evaluation techniques and the modification of several old ones. The first part of this paper describes the apparatus, procedures and their valuations and the second part gives the results from the survey. The original survey as reported in this paper is

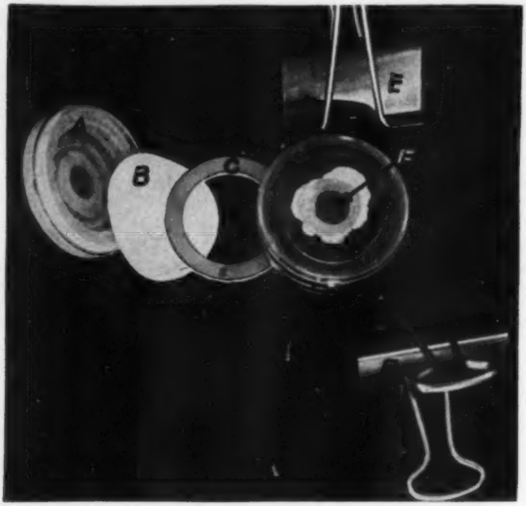


FIG. 1—EXPLODED VIEW OF PLASTIC MILLIPORE FILTER HOLDER

being extended to include a study of the loading characteristics of representative types of filters in typical service.

SAMPLING EQUIPMENT

Pumps: To permit sampling in occupied spaces the small, quiet and inexpensive diaphragm pump used in the *AISI* smoke samplers were used. These pumps were mounted in small wooden boxes lined with glass fiber sound insulating material. Leakage around the actuator gland required monthly calibration against a wet test meter to obtain a ± 5 percent accuracy of the estimated air volume sampled.

Filter Holders: The large number of filter holders required led to the development of the inexpensive plastic holders shown in Fig. 1. Designed for the standard unmounted 47 mm size millipore filters, these holders are made from commercially available 47 mm size plastic petri dishes.

Part A consists of the bottom of a petri dish with a ½-in. copper tube hose connection cemented in place and a 40-mesh supporting screen cemented to a 47-mm

plastic ring (also available commercially). The ring and screen are cemented in place in the bottom of the dish. B is the millipore filter; C is a gasket ring cut from laminated sponge rubber and leather shoe innersole; D consists of a petri dish top and bottom cemented together with a large hole cut through; F is a dish top with a piece of 170-mesh stainless steel lint screen cemented in place over a ½-in. hole, and E shows paper clips used to hold the assembly together.

A sampling pump and holders assembled for simultaneous indoor and outdoor sampling is shown in Fig. 2.

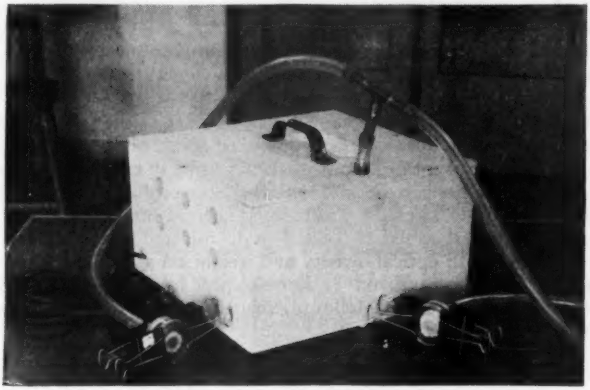


FIG. 2—SAMPLING PUMP AND FILTER HOLDERS ASSEMBLED FOR SIMULTANEOUS INDOOR AND OUTDOOR SAMPLING

Dust Fall Slides: *Standard* 25 × 75 mm microscope slides resting on small wood blocks were used for dust fall sampling. Before use these slides were cleaned carefully with detergent and water.

SAMPLING PROCEDURES

Sampling Locations: Sampling locations were determined by a number of factors. Participation was limited to the University of Minnesota and 4 members of the Research Technical Advisory Committee on Air Cleaning of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS. Locations in any given city were limited to those readily accessible to the sponsor. Though the sampling pumps were reasonably quiet, locations had to be found where the owners or occupants were willing to tolerate the occasional nuisance of a noisy pump and the weekly visits to change sampling media. Thus the number and location of samples taken represents the best compromise that could be made under the circumstances. Specific locations are discussed with the results.

Typical Sampling Procedure: The light transmissions of the clean millipore filters were read before use. The assembled samplers were then placed in opera-

tion with the dust fall slide located nearby. Sampling periods varied from 2 days to a week depending on location, dust concentration and the number of filter holders attached to a pump. The lint screen was omitted by those participants not having a photometer suitable for the evaluation of the lint screen.

SAMPLE EVALUATION

Dust Spot: Optical density of the loaded millipore filters was determined with the improved photometer described previously[1] at a 0.75-in. diaphragm opening. These measured optical densities were then used to calculate the staining dust concentration in cohs and for comparison with the optical densities calculated from the sedimentation analysis.

Lint Screen: The pronounced effects of fibrous particles on ventilation filter life made it desirable to incorporate a measure of the concentration of such particles in the survey.

After considerable study the photometric evaluation of the particles caught on a 1.27 sq cm 170-mesh stainless steel screen placed ahead of the millipore filter was adopted. The small amount of fibrous material normally collected precluded evaluation by weight. Lint screen samples were therefore evaluated with the improved photometer[1]. The optical density was calculated from measurements of the light transmission through the screen with the dust deposit and after thorough washing.

Measurements of both optical density and weight on a special 8 cm diam, 170-mesh screen by Engebretson[2], on 11 lint samples from typical residential and non-residential locations gave the following relationship between weight and screen light transmission:

$$\text{mg/cm}^2 = 2.52 \log (I_o/I) \quad \ldots \ldots \ldots \ldots \quad (1)$$

The linear correlation was 0.989 which is surprisingly good. Lint concentrations given in the latter part of this paper are calculated using Equation 1. Not all particles collected by the screen are fibrous. Detailed study of mechanism of collection of the screen by Engebretson[2] has shown that inertial impaction of non-fibrous particles above approximately 5 μ in size can account for a substantial fraction of the total collected. Observations on particles collected on the screens during the survey indicate from 50 to 90 percent of the total can be considered as fibrous. It is, therefore, believed that this simple lint screen sampling method provides a rough but useful index of the proportion of fibrous particles normally found in airborne dust.

Dust Fall: Dust fall measurement by the use of jars placed out of doors has been standard for many years. Microscope evaluation of dust fall on microscope slides has been used by the telephone company. More recently Hemeon[3] has used photomultiplier measurement of light scattered by dust deposits on microscope slides. Though very sensitive this latter technique suffers because of the difficulty of calibration.

For the ASHAE survey a simple rapid method of moderate accuracy was obtained by increasing the sensitivity of the transmission photometer used for dust spot measurements to the point where the deposits could be evaluated by reduction in the light transmission through the slide. Sensitivity was sufficient to measure

---

[1]Exponent numerals refer to References.

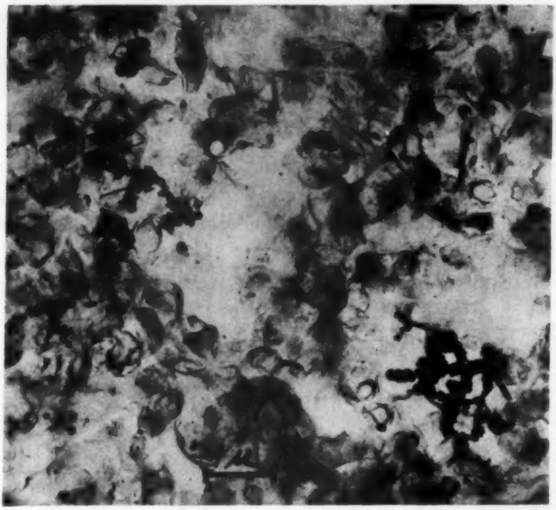


FIG. 3(A)—ELECTRON PHOTOMICROGRAPH OF MILLIPORE COLLECTED AIR-BORNE DUST, SAMPLE OC-1, MEASURED OD = 0.046

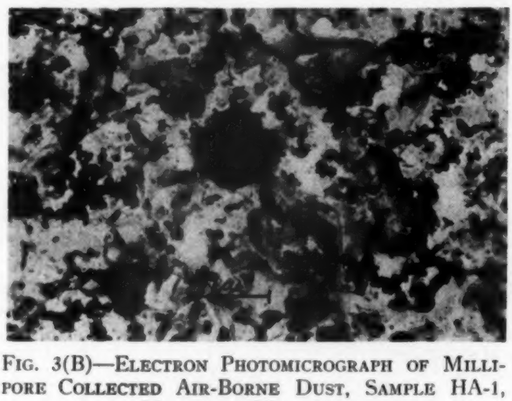


FIG. 3(B)—ELECTRON PHOTOMICROGRAPH OF MILLIPORE COLLECTED AIR-BORNE DUST, SAMPLE HA-1, MEASURED OD = 0.094

the dust fall accumulated in one day in a clean location with an accuracy of about ± 20 percent.

Since the photometer measures optical density it has been found convenient to define a unit rate of dust fall in terms of optical density. This unit rate has been designated, for convenience, as *O.D.D.* and is defined as that quantity of dust fall per day which has an optical density of 0.001, thus

$$1\ O.D.D. = 10^{-3} \log (I_o/I) \quad (2)$$

An evaluation of this dust fall measurement technique by Engebretson[2] showed that the *O.D.D.*—weight per unit area relationship depends somewhat on the angle

TABLE 1—COMPARISON OF MEASURED AND CALCULATED OPTICAL DENSITIES FOR ATMOSPHERIC DUST SAMPLES HA-1 AND HA-3

Distribution	Measured *OD*	Calculated	
		$K = 1$	*K* for Diff Theory[a]
Sample HA-1			
Light Microscope Alone	0.094	0.0106	0.0164
Electron Microscope Alone	0.094	0.0586	0.0654
Sedimentation Alone	0.417	0.290	0.437
Combined	0.094	0.0706	0.0914
Sample HA-3			
Light Microscope Alone	0.122	0.0155	0.0187
Electron Microscope Alone	0.122	0.1008	0.0128
Sedimentation Alone	0.568	0.324	0.460
Combined	0.122	0.115	0.104

[a] The light scattering coefficient, *K*, used is that suggested by Davies[4] for opaque particles in a transparent medium. No satisfactory data exist for opaque particles on a translucent filter medium.

of acceptance of the measuring photocell. At a semi-angle of acceptance of 16 deg, evaluation of samples from 6 representative locations gave the following relationship. (Variance was 7 percent).

$$\log_{10} (I_o/I) = OD^* = 1.88 \text{ mg/cm}^2 \quad (3)$$

$$1\ O.D.D. = 1.66 \text{ tons per square mile per month} \quad (4)$$

Particle Size Analysis: Because of the extremely broad size range and variable character of air-borne dusts there exists no single size analysis method which can be used to obtain satisfactory accuracy over the entire range from about 0.02 to 50 μ. For this study it was desired to measure or calculate the volume and area size distributions. The volume distribution was used because the volume of air-borne dust determines filter life and the area distribution because this is related to the soiling power of the dust[4].

*Throughout the paper *OD* is used to represent optical density.

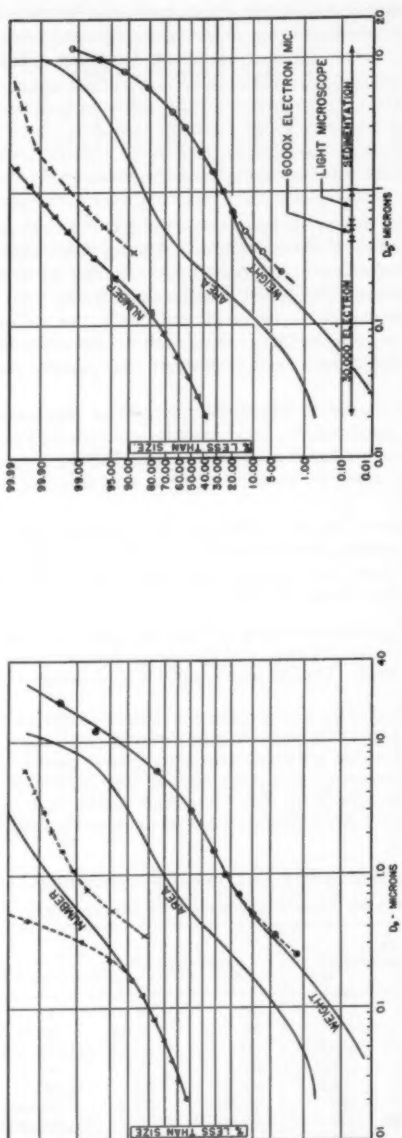


FIG. 4—TYPICAL AIR-BORNE SIZE DISTRIBUTIONS FROM COMBINED LIGHT AND ELECTRON MICROSCOPE AND SEDIMENTATION DATA. DELTAS REPRESENT MEASUREMENT BY ELECTRON MICROSCOPE; CROSSES REPRESENT MEASUREMENTS BY LIGHT MICROSCOPE, AND CIRCLES SHOW MEASUREMENT BY SEDIMENTATION

TABLE 2—SEDIMENT COLUMN POROSITIES FOR SEVERAL DUSTS

DUST	D_{gs}	NO. OF SAMPLES	ASSUMED DENSITY	POROSITY θ	S, %
Atmospheric	3.0	13	2.1	0.67	54
Arizona Road Dust	2.0	8	2.7	0.55	
Precipitated Carbon Black	4.5	6	2.1	0.88	

Choice of the centrifuge sedimentation analysis method[5] as the primary method was determined by a number of considerations. Among these were availability of the equipment, speed, convenience and reasonable accuracy over the 0.3 to 25 μ size range of primary interest. A limited number of analyses were made with the light and electron microscope to supplement and check the sedimentation data. Also a few comparisons of sedimentation data with results obtained by other investigators with the jet impactor, particle counter and parallel plate air settling chamber were obtained.

Light Microscope Size Analysis: The light microscope size analysis procedure used has been described previously[4]. Dust samples are collected on hydrosol-type millipore filters and are counted and measured using a Porton-type reticule at high magnification. The equivalent of 10,000 to 15,000 was counted by a multistage procedure.

Electron Microscope Size Analysis: Through the courtesy of J. Sayer and M. Jacoby, a procedure for taking good electron microphotographs of millipore filter collected samples of air-borne dust was developed and several samples thoroughly analyzed. The procedure is briefly as follows:

A 500 to 10,000 A° carbon film is evaporated onto the surface of the millipore filter. The millipore filter is then dissolved away and the carbon film containing the embedded particles is deposited on a grid. Typical photographs of dust sampled in the laboratory are shown in Fig. 3.

Multistage counts on from 3 to 5 photographs enlarged 12,000 to 30,000 times are then made by comparison with black circles having diameter ratios equal to the $\sqrt{2}$. To increase the probability of seeing the larger particles without resorting to an excessive number of photographs, counts on millipores having both high (0.4) and low (0.04) optical densities or on photographs at different magnification were used. The ratios of the measured optical densities can then be used to combine the counts.

TABLE 3—COMPOSITE SIZE ANALYSES OF ATMOSPHERIC DUST

(Sampled Inside University of Minnesota Particle Laboratory)

PARTICLE SIZE MICRONS (μ)	OC-1 MARCH 1956		HA-1 JANUARY 1957		HA-3 JANUARY 1957	
	NO.	WT	NO.	WT	NO.	WT
0.02			52	0.006	38	0.008
0.04	0.7		64	0.02	52	0.03
0.1	7.5	0.02	80	0.20	75	0.5
0.4	87	5.6	97.5	7.1	98.9	13.6
1	99	15.0	99.8	23.1	99.94	26
4	99 98	60	99.995	59.5	99.998	66
10	99.9996	90	99.99999	92.5	99.99998	97.0
20		99		99.7		100.0

Because of the limited number of size analyses made using the electron microscope, only limited conclusions as to its accuracy and usefulness can be made. The time-consuming nature of the count procedures and severe sampling problems probably make it mandatory to combine the electron microscope data with light microscope or preferably sedimentation analysis if misleading results are to be avoided. The good agreement between calculated and measured *OD's* shown in Table 1 indicates that the carbon film replication technique does account for most of the particles. However further comparisons with thermally precipitated dust samples are needed to accurately determine the loss of very fine particles (0.01 to 0.1μ).

Centrifuge Sedimentation Analysis Procedure: The procedure used is essentially that described previously[2,4,5]. It is briefly as follows. A millipore filter containing between 0.3 and 1.0 mg of dust is dissolved in acetone in a 40 ml centrifuge tube

TABLE 4—AVERAGE DUST PROPERTIES

MEDIAN SIZE INSIDE RESIDENCES

	No. Samples	D_{gs}-μ	$\bar{S}$-%
All residential	26	3.17	24
All Minneapolis residential	17	3.31	23
South Minneapolis	13	3.41	17
Southeast Mpls. Apt.	4	3.00	38
Louisville residential	6	2.64	27
Pittsburgh residential	3	3.57	9

MEDIAN SIZE INSIDE LABS AND OFFICES

	No. Samples	D_{gs}-μ	S-%
All samples	67	3.01	29
Minneapolis Labs & Offices	64	2.69	37
Minneapolis Labs	41	2.35	35
Minneapolis Offices	24	3.30	31
Louisville Labs & Offices	3	3.32	16

MEDIAN SIZE—OUTSIDE AIR

	No. Samples	D_{gs}-μ	S-%
All samples	40	3.56	33
Minneapolis (all)	23	4.31	28
Minneapolis Lab	13	4.09	30
Minneapolis, south residences	8	4.95	20
Louisville (all)	9	2.65	21
Louisville Lab & Office	3	3.10	21
Louisville Residential	6	2.43	13
Akron (Residences & Lab)	5	2.82	10
Pittsburgh	3	4.47	22

using moderate stirring. The suspension is then centrifuged at 3000 rpm for 1 hr to precipitate the particles, and the supernatant liquid syphoned off. The dust is resuspended in 1 ml of feeding liquid consisting of approximately 15 percent naphtha—85 percent acetone. This suspension is then floated as a layer on a pure acetone sedimentation liquid and the size analysis carried out by the previously described techniques.

Evaluation of the Centrifuge Sedimentation Procedure: Because of the use of a sedimentation method for the analysis of air-borne dusts may be seriously questioned, a rather thorough evaluation of its limitations and probable accuracy was made. The most important considerations and a summary of their evaluations are as follows:

1. *Dispersion.* The question arises as to whether the sedimentation particle size corresponds to that existing in the air. Throughout the survey the measured optical densities of the loaded millipore filters were compared with the optical density calculated from the sedimentation size distribution and sediment volume by the method described

NOMENCLATURE

I_o = light transmission of a clean filter medium.
I = light transmission of a dirty filter medium.
OD = $\log_{10} (I_o/I)$.
$O.D.D.$ = $[112 (I_o\text{-}I]/t$ (thousandths of an OD of dust fall per day).
t = time in hours.
θ = porosity.
D_p = particle diameter, microns.
D_{g1} = number geometric mean size, microns.
D_{g3} = volume geometric mean size, microns.
S = variance.
C = concentration, grains per 1000 cu ft.
K = light scattering coefficient = $\dfrac{\text{effective scattering area}}{\text{true particle projected area}}$.
μ = micron.

earlier[4]. Details of this comparison are given later. The overall ratio of measured to calculated OD was 1.93. It is believed that perhaps one-half of the difference between 1 and 1.93 is due to the use of an incorrect K value in the computations. The indications from the detailed studies of the measured optical densities of particles on a millipore at different angles of acceptance by Engebretson[2], from some relatively unsuccessful attempts at measuring the K value directly and from comparisons of measured and calculated OD's for combined size analysis such as Fig. 4, are that the K values are higher at larger particle sizes than is predicted by the diffraction theory curve actually used.

It is therefore believed that the observed agreement between calculated and measured OD's indicates sedimentation above 0.3 μ is essentially the same as that existing on the millipore except possibly for very large and fragile aggregates.

2. *Millipore dust load.* It was found that proper dispersion could not be obtained if the millipore filter contained more than about 1 mg of normal air-borne dust. Irreversible agglomeration of the dust on the millipore apparently takes place if an appreciable number of particles touch each other in the dry state. From circumstantial evidence accumulated during the survey, it appears that normal air-borne dust particles are often coated with sticky materials which polymerize in the air stream after the particles are deposited on the filter. Predominently mineral dusts do not exhibit this irreversible behavior.

During the survey dust loads were normally kept below 0.5 mg per 47 mm diam millipore.

3. *Acetone soluble particles.* The sedimentation size analysis techniques employed measure only the acetone insoluble particles. The work of Chambers et al[6] indicates that the average city dust probably contains 15 to 20 percent acetone soluble particles. It is, therefore, probable that the dust concentrations computed from the sediment heights are about 15 percent low on the average. The size distributions reported are therefore only those of the acetone insoluble fraction, since all methods of size analysis used destroy the soluble particles before measurement.

4. *Sediment column bulk density.* Computations of the dust concentration depend on the sediment column bulk density which in turn is a function of the true particle density and the sediment porosity.

Acetone displacement measurement of the density of 5 samples of dust taken from electronic air cleaners operating dry in representative locations gave densities ranging from 1.74 to 2.43 gm per cc with a mean of 2.06. The lowest value was obtained on dust taken from a residential air cleaner. A density value of 2.1 gm per cc was assumed for all calculations reported in this paper.

Measurements of the porosity of the sediment column in the centrifuge tube were made on 13 samples of atmospheric dust by weighing millipore samples and then determining the sediment height. To obtain greater weighing accuracy the sediment from a number of the millipore samples was washed from the centrifuge tubes and reweighed in tared aluminum dishes. Results are given in Table 2. A value of 0.6 was used for all computations reported in this paper.

For atmospheric dust, values of θ (where θ = porosity) ranged from 0.4 to 0.84 with the highest values being determined on dust from residential samples or from laboratory samples containing a large proportion of carbon aggregates.

5. *Sediment column compaction.* Some compaction was experienced with samples containing more than a few percent of fibrous particles. Samples taken without a lint screen preceding the millipore filter were poured through a 325-mesh screen while suspended in the feeding liquid in order to eliminate fibrous particles. Thus the sedimentation data reported in this paper is for the minus 325-mesh fraction only.

The effect of compaction during a size analysis is to give a low estimate of the amount of particles below approximately 0.8 μ This can be seen in Fig. 4.

6. *Reproducibility.* The variance of the size analysis on 4 simultaneous samples was determined by Engebretson[2] to be: S = 16 percent at 5th percentile, 9 percent at 50th percentile, and 17 percent at the 95th percentile.

Combined Size Analysis: As a check on the sedimentation size analysis method, 3 complete analyses of millipore dust samples collected in the laboratory were made, using combined light microscope, electron microscope and sedimentation data. Results of these 3 analyses are tabulated in Table 3 and the latter 2 plotted in Fig. 4.

These distributions were computed† using light and electron microscope data below 1 μ and sedimentation data just mentioned. Upper size limits for the light and electron microscope were taken equal to the size where the probability of seeing a larger particle is about equal to 10^{-4}. This limit was chosen because it is impractical to count more than about 10^4 particles even with multistage procedures. Lower limit for the light microscope is determined by the limit of resolution.

The lower limit for the sedimentation analysis, as used in the combined analysis, is 1 μ. Actually the useful lower limit is about equal to the size at which the cumu-

†Due to the involved nature of this work, it is planned to write a separate paper describing the method of analysis, theory and computational procedures.

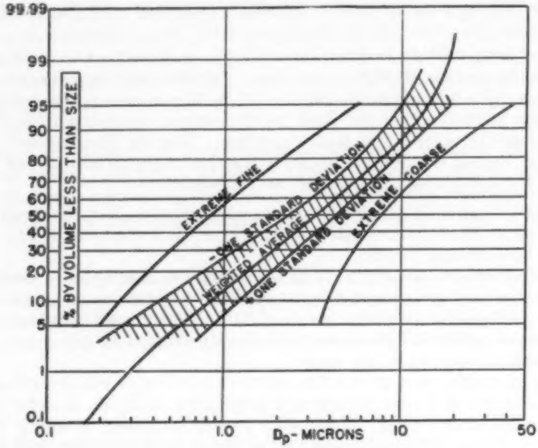


FIG. 5—SEDIMENTATION SIZE DISTRIBUTION OF AIR-BORNE DUST WET-SIEVED THROUGH 325-MESH SCREENING

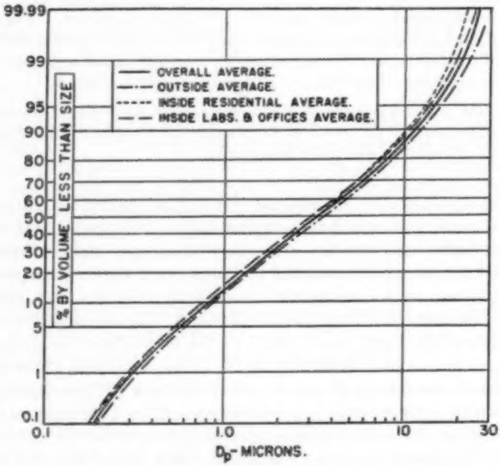


FIG. 6—AVERAGE SEDIMENTATION SIZE DISTRIBUTIONS WET-SIEVED THROUGH 325-MESH SCREENING FOR THREE LOCATIONS

lative percent is about equal to the sediment column reading accuracy. For normal air-borne dust this is between 0.2 and 0.4 μ.

As a further check on the accuracy of the measured size distributions, the optical densities for the individual and combined distributions were calculated and compared with the measured *OD's*. These are tabulated in Table 1.

From Fig. 4 and Tables 3 and 1, it is possible to draw a number of conclusions.

1. The light microscope can at best give only a poor estimate of the complete size distribution of normal air-borne dust. This is shown by the poor agreement between

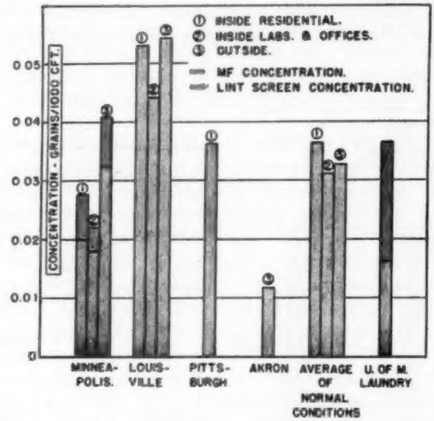


FIG. 7—DUST CONCENTRATIONS FOR SEVERAL LOCATIONS

calculated and measured *OD's* in Table 1 and the fact that only 0.2 to 0.1 percent or less of the particles can be seen using practical techniques.

2. A single-stage electron microscope count such as was used on sample HA-1 is inadequate. However, the multi-stage E M count used on HA-3 (Fig. 4B) appears to give accurate data up to about 1 μ.

3. The sedimentation method will be most accurate above 1 μ but gives a progressively lower estimate below 1 μ. From Fig. 4 it can be seen this error is not too serious above about 0.4 μ.

4. Probably the most accurate method would be a combination of sedimentation and electron microscope.

5. From Fig. 4B, it can be noted that the 99.9 percent by number of particles below 1 μ accounts for over 90 percent of the projected area or staining power, but only about 25 percent of the total volume. This illustrates clearly that the rating of air cleaners by number, dust spot or weight arrestance have quite different meanings.

DUST SURVEY RESULTS

Sampling Frequency and Location: Dust survey samples were taken by 6 contributing groups in 4 individual cities; *viz*: 3 in Pittsburgh, one in Louisville, one in

Akron, and the University of Minnesota in Minneapolis. The majority of sampling was done in Minneapolis and Louisville, but enough samples were received from the other cities to indicate the amount of variation to be expected between extreme conditions.

The groups took samples in the following locations: inside and outside residences, inside and outside labs and offices, inside industrial plants. The inside locations represented a range of conditions from old downtown-area buildings employing no air cleaning to newly-remodeled, windowless offices with modern air cleaning equipment. Outside sampling was done in areas ranging from commercial districts to clean residential areas.

Unfortunately, it was impossible to maintain continuous or nearly-continuous sampling at any given location throughout a complete year's cycle. Only at the

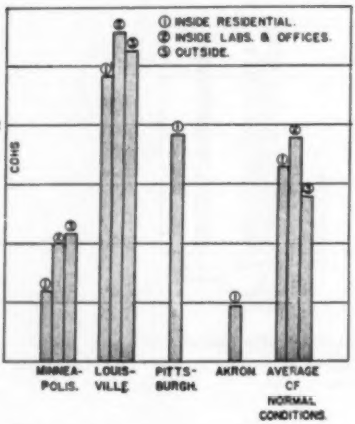


Fig. 8—Staining Concentration

SHAPE	APPEAR-ANCE	KINDS	% IN DUST BY WT. RANGE	AVE.
SPHERICAL		SMOKES POLLEN FLY ASH	0-20	10
IRREGULAR CUBICAL		MINERAL CINDER	10-90	40
FLAKES		MINERALS EPIDERMIS	0-10	5
FIBROUS		LINT PLANT FIBER	3-35	10
CONDENSA-TION FLOCCS		CARBON SMOKES FUMES	0-40	15

Fig. 9—Air-Borne Dust Particle Shapes

University of Minnesota were enough samples obtained to test for seasonal variations.

Averaging Procedure: All *overall* average values are the result of weighting equally the 3 major location categories: (1) inside residences, (2) inside labs and offices, and (3) outside. This gives twice as much weight to inside locations as to outside, which might be justified on the grounds that a high percentage of the air passing through most ventilation air cleaning systems is recirculated air, but the average values at each location are so similar that any reasonable system of weighting would produce essentially the same results.

The average values *inside residences* were obtained by weighting equally the averages for 3 cities—Louisville, Minneapolis, and Pittsburgh. *Inside labs and offices* averages were computed by giving equal weight to the average values for Minneapolis and Louisville. The *outside* average values gave equal weight to averages from 3 cities—Akron, Louisville and Minneapolis—on all items except those

pertaining to size distribution. For the 5, 50, and 95 percentile sizes, averages from Pittsburgh were included at equal weight.

Average Dust Properties: The *geometric mean size* by volume (D_{g3}) of natural air-borne dust was found to vary with location, and displayed a range of values at any given location. The mean size was 3.25 μ, with a 30 percent variance (S). Inside residences, D_{g3} was 3.17 μ (S = 24 percent).

A breakdown of these data is contained in Table 4. The south Minneapolis samples were obtained in 2 separate, but closely-located homes in the extreme southern part of the city. The Louisville sampling was done in 3 homes. The overall average of D_{g3} = 3.17 ± 24 percent covers a range which includes all of the individual averages shown.

Inside labs and offices, D_{g3} = 3.01 μ (S = 29 percent). Table 3 contains the more detailed data from which this average was obtained. Offices and laboratories were combined as one category because of the similarity of their construction and atmospheres. Both are relatively lint-free and clean, and do not have direct outside entrances.

Outside air had a geometric mean size by volume of D_{g3} = 3.56 μ (S = 33 percent). Table 4 shows the detailed data. Louisville mean size of 2.65 μ, it will be noted, was considerably less than Minneapolis average mean of 4.31 μ, and even further from the south Minnesapolis residential mean of 4.95 μ.

As noted from the variance, the geometric mean size has considerable spread. The values for inside residences ranged from 1.34 to 4.60 μ and for inside labs and offices from 0.83 to 5.15 μ. For outside air the range was 2.69 to 7.60 μ.

At a confidence level of 95 percent or higher, the average inside residential and inside labs and offices means are equal, but each of these varies significantly from the outside average.

In general, *the cumulative size distribution curve* is bimodal. An overall average curve, weighting equally the 3 major classes of location, is shown in Fig. 5 with 5 percent, by weight, smaller than 0.56 μ, 50 percent smaller than 3.25 μ, and 95 percent smaller than 14.3 μ. Fig. 5 includes, additionally, the curves corresponding to plus and minus one standard deviation, and the extreme distributions found. The distributions for each of the 3 major locations, shown in Fig. 6, are very similar to the overall average distribution and warrant no further explanation. The size distributions for individual samples were strongly bimodal, but this characteristic was diminished by successive average processes.

Because all of the groups were not equipped to measure *the lint concentrations*, an overall average is not possible. Considerable information is available, however, from the Minneapolis data. Inside Minneapolis residences, the lint screen loading constituted an average of 27.6 percent of the total dust weight. Breaking this average down further, 13 south Minneapolis samples showed an average 27.8 percent and 4 southeast Minneapolis samples had an average of 27.4 percent. Dust samples taken inside Minneapolis labs and offices had an average lint screen loading of 17.2 percent, by weight. Considering the 24 office samples alone, offices had 16.6 percent. The 41 lab samples had an average of 18.0 percent. Air outside the Minneapolis lab contained 21.5 percent and air outside south Minneapolis residences contained 16.4 percent.

Average Concentrations: The air-borne dust retained on the millipore filter had an overall average concentration of C = 0.0335 grains per thousand cu ft (S = 77 percent). No overall lint-screen concentration is possible for reasons explained previously. Fig. 7 illustrates the differences in concentrations for the 3 classes of location between cities, plus information on an unusual location.

The average MF (millipore filter) concentration *inside residences* was $C = 0.0365$ gr per tcf* ($S = 85$ percent). Minneapolis residences had an average MF concentration of 0.0200 gr per tcf ($S = 67$ percent) and a lint concentration of 0.0076 gr per tcf ($S = 80$ percent) for a total of 0.0276 gr per tcf ($S = 61$ percent). The south Minneapolis residential samples, taken alone, averaged 0.0162 gr per tcf MF dust ($S = 25$ percent), 0.0062 gr per tcf lint ($S = 57$ percent) and 0.0244 gr per tcf total ($S = 34$ percent). Louisville residences had an average MF concentration of 0.0531 gr per tcf ($S = 86$ percent) while the Pittsburgh MF concentration was 0.0364 gr per tcf ($S = 21$ percent).

The average MF concentration *inside labs and offices* was 0.312 gr per tcf ($S = 70$ percent). The Louisville MF samples averaged 0.0442 gr per tcf ($S = 53$ percent) while the Minneapolis MF average was 0.0183 gr per tcf ($S = 52$ percent). In Minneapolis an additional 0.0038 gr per tcf ($S = 89$ percent) was measured as lint, for a total concentration of 0.0221 gr per tcf ($S = 56$ percent).

TABLE 5—DUST FALL IN MINNEAPOLIS

LOCATION	NO. OF SAMPLES	O.D.D.	S-%
Inside Labs & Offices	30	0.80	66
Inside Labs	8	0.63	91
Inside Offices	22	0.86	57
Inside Residences	12	0.93	56
Outside	13	4.92	26

Samples of *outside air* had an average MF concentration of 0.0328 gr per tcf ($S = 71$ percent). Akron MF concentrations averaged 0.0118 gr per tcf ($S = 45$ percent) while Louisville had 0.0543 gr per tcf ($S = 42$ percent) and Minneapolis had 0.0324 gr per tcf ($S = 42$ percent). In addition, Minneapolis had a lint concentration of 0.0085 gr per tcf ($S = 59$ percent) for a total concentration of 0.0409 gr per tcf ($S = 27$ percent).

Measured Cohs: The measured coh values for the principal locations and cities are as shown in Fig. 8. The most significant fact is that the variation between cities is much more pronounced than the variation between locations within any one city. The locational variations within each city are of doubtful significance.

Ratio of Measured Cohs to Calculated Cohs: The overall average ratio of measured to calculated cohs was 1.93 ($S = 46$ percent), corresponding to a mean size of 3.25 μ. The inside residential ratio was 1.82 ($S = 48$ percent), corresponding to $D_{g3} = 3.17\ \mu$, while the value inside labs and offices was 2.19 ($S = 47$ percent) for $D_{g3} = 3.56\ \mu$.

As a further check on the effect of size distribution on the ratio of measured to calculated cohs, the average ratio of all samples with a mean size less than 2 μ was compared to the average ratio for samples with mean diameters greater than 4 μ. The *under 2-μ* samples yielded an average $D_{g3} = 1.58\ \mu$ and a ratio of measured to calculated cohs of 1.43. The *over 4-μ* samples had a D_{g3} of 4.83 μ and a ratio of

*gr per tcf is grains per thousand cubic feet.

2.68. The difference between these ratios was significant at a 99.9 percent confidence level.

Dust Fall: Measurements of the atmospheric dust fall were made only at Minneapolis. The respective values are given in Table 5.

The variation between labs and offices was not significant at a 95 percent confidence level, and similarly the variation between *residences* and *labs and offices* was insignificant. However, the average value for outside samples is significantly different from average of all inside samples. Therefore, it may be concluded that in Minneapolis the dust fall is essentially constant for all normal inside locations, although scattered data indicates significant variations between rooms of a particular house.

TABLE 6—PROPERTIES OF AVERAGE AIR-BORNE DUST

PROPERTY	RANGE	AVERAGE
Weight[a] geometric mean size = D_{g3}	0.8 – 8	3 μ
Number geometric size = D_{g1}	– to 0.2	0.03 μ
Particle density	1.5 – 3.0	2.1 gm/cc
Porosity =	0.85 – 0.05	0.65
Bulk density	0.2 – 1.5	0.7 gm/cc
Fraction fibrous particles[b]	3 – 35	10%
Upper particle size[a]	15 – 50	25 μ
Terminal velocity of largest particles	2 – 10	3 cm/sec
Dust fall	0.4 – 10	1.5 *O.D.D.*
Size distributions	(Figs. 4 & 5)	
Concentration	(Figs. 7 & 9)	
Physical shape of particles	(Fig. 9)	

[a] This is the Stoke's equivalent spherical size calculated from settling velocity. Large irregular particles of low density such as lint may be larger physically in one or more dimensions.
[b] The values given here represent the actual fraction of particles that can be considered fibrous. This will be from 50 to 90 percent of the weight of dust that can be caught on the 170-mesh screens used on the millipore samplers.

Seasonal Variations: A significant seasonal variation in mean dust size was evidenced in samples taken inside the Minneapolis lab, the only sampling location having sufficient year-round data for such an analysis. The year was divided into 2 periods: (1) the non-heating season from June 1 to August 31, and (2) the remainder of the year, September 1 to May 31. The average for the period June 1 to August 31 was $D_{g3} = 1.83\ \mu$ while the remainder of the year had an average of $2.58\ \mu$. This difference in mean sizes was significant at a 95 percent confidence level.

Significant variations were also noted in millipore filter and lint screen concentrations, for the same location. The MF concentration for the period, June 1 to August 31, averaged 0.0130 gr per tcf (S = 25 percent), while for the remainder of the year it averaged 0.0175 gr per tcf (S = 34 percent). This difference was significant at a 95 percent confidence level. The corresponding lint screen concentrations were 0.0022 gr per tcf (S = 65 percent) and 0.0042 gr per tcf (S = 42 percent), respectively. This difference was significant at better than a 99 percent level.

The bimodal character of the size distribution, referred to previously, was appreciably diminished or nonexistent within experimental accuracy in the majority of summer samples. The summer atmospheric dust, it has been noted has finer distribution, a lower concentration, and a measured to calculated coh ratio closer

to unity than the dust existing during the remainder of the year. This may be explained by considering the normal atmospheric dust to consist of 2 general types of material, besides the lint; a mineral fraction of relatively fine size distribution, and a carbonaceous fraction of relatively coarse distribution. These 2 overlapping distributions would give the bimodal characteristic in the fall, winter, and spring, with the carbonaceous fraction gradually diminishing until an almost unimodal distribution is obtained during the summer. The mineral fraction concentration is believed to remain fairly constant year-round. The lower measured to calculated coh ratio substantiates this theory, because supplementary work has indicated that the principal error in the K values assumed for the purpose of this work, is at the coarse end of the range. Additional evidence is the fact that the summer samples are brown rather than black.

CONCLUSIONS

Following are the important conclusions from this survey and from the analysis of work by other investigators. Because of the limited number of samples and sampling locations in the present survey, the numerical values in Table 6 represent a combination of straightforward statistical weighting and judgment based on less tangible factors. It should be emphasized that, to a certain extent, results are dependent on the analysis methods used.

Variation in Properties and Concentration

1. The differences in particle character and concentration between clean non-industrial cities such as Minneapolis and polluted industrial cities are much greater than the differences in size distribution.

2. The variations in normal air-borne dust are large enough to make it suitable as a test dust only for rough performance testing of air cleaners.

3. For Minneapolis, summer dust is significantly finer, lighter colored, of lower concentration and less bi-modal in size distribution than dust during the rest of the year. This may be attributed to a reduction in the carbonaceous and fly ash particles during the summer.

4. The large differences in the amount of fibrous particles found in different locations may have very significant effects on the operation of air cleaners. This conclusion is substantiated by current research on filter life.

Miscellaneous Conclusions

1. In normal air-borne dust most staining is done by particles less than 5 μ in size.

2. Most air-borne dust aggregates are tightly bonded together and require severe mechanical dispersion to break them up.

3. Natural air-borne dust particles collected in bulk suffer irreversible agglomeration due to polymerization of compounds in the dust.

4. The millipore filter sampling and sedimentation weight concentration and size analysis technique is convenient, rapid and flexible. It appears to be well suited to general purpose air-borne dust evaluation.

REFERENCES

1. ASHAE RESEARCH REPORT No. 1597—The Dust Spot Method for Evaluating Air Cleaners, by K. T. Whitby, A. B. Algren and R. C. Jordan (ASHAE TRANSACTIONS, Vol. 63, 1957).

2. A Study of Sampling and Measuring Techniques for Atmospheric Dust, by W. E. Engebretson (*Thesis*, University of Minnesota, September 1956).

3. Unpublished work at the Mellon Institute by W. C. L. Hemeon, Hemeon Associates, Pittsburgh.

4. ASHAE RESEARCH REPORT No. 1549—Size Distribution and Concentration of Air-borne Dust, by K. T. Whitby, A. B. Algren and R. C. Jordan (ASHAE TRANSACTIONS, Vol. 61, 1955, pp. 463-483).

5. A Rapid General Purpose Centrifuge Sedimentation Method for Measurement of Size Distribution of Small Particles: ASHAE RESEARCH REPORT No. 1525—Part I—Apparatus and Method, and ASHAE RESEARCH REPORT No. 1548—Part II—Procedures and Applications, by K. T. Whitby (ASHAE TRANSACTIONS, Vol. 61, 1955, pp. 33-50 and 449-462).

6. The Characteristics and Distribution of Organic Substances in the Air of Some American Cities, by L. A. Chambers, E. C. Tabor and M. J. Foter (*R. A. Taft Sanitary Engineering Center, U. S. Public Health Service*, Cincinnati, Ohio, May 1955).

DISCUSSION

R. F. LOGSDON, Louisville, Ky. (WRITTEN): The authors have carried out a very important investigation, its importance resting mainly on its influence in establishing an aerosol for air filter evaluation. As a representative of one of the cooperating laboratories I feel free to comment on the limited scope of this survey. I think one of the primary objectives of the ASHAE Air Cleaning Committee is to obtain basic knowledge on the physical characteristics of air-borne particulate and to relate these characteristics and properties to air cleaner performance. With this in mind it is regrettable that the survey could not have been more extensive. Seasonal variation, for example was checked at only one station, Minneapolis. The differences noted there in both mean particle size and lint content point up the need for additional information on seasonal effects. Lint samples are also very limited, and this very important factor varied from 3 percent to 35 percent in the samples that were taken. These extremes suggest that additional sampling is desirable to better establish the fibrous content of atmospheric particulate.

The authors refer to the filter loading survey in terms of a continuation of this dust survey. No doubt the filter loading survey will pick up some of the loose ends of the work described in this paper and may invalidate to some extent my comments regarding the limitations of this work.

In my opinion the dust survey described here, and the filter loading survey not yet completed, can and will represent the most important contribution to the Society's air cleaning program. I urge that they be carried out as extensively as possible and, further, I feel strongly that the results of both surveys, undertaken to obtain basic information, should be known and published before any of the more specific details of filter test procedure or equipment are described. This would appear to be a logical sequence since procedure and equipment will be influenced greatly by the findings of the surveys.

Regarding specific points in the author's paper, I wish to make the following comments.

1. The acetone soluble particles may represent about 15 to 20 percent (and could be considerably more or less) of the total particulate in a sample. As the authors state, this soluble fraction does not show up in the particle size analysis. This is too great an amount to ignore. Additional work is indicated as there is no reason to believe the soluble fraction would not affect the distribution curve.

2. The irreversible agglomeration phenomenon described by the authors may be affected by the solution of the millipore paper in acetone. Our experience has indicated that the resulting resinous mixture has adhesive characteristics.

3. Conclusion 2 states "the variations in normal air-borne dust are large enough to make it suitable as a test dust only for rough performance testing of air cleaners".

I agree especially when testing on a weight basis. Tests on a dust-spot or discoloration basis, however, using normal air-borne particulate, will yield more consistent results as the discoloration method is more critical to fine particles. Size distribution on a weight basis could vary considerably without greatly affecting the staining power. This is brought out also by the authors who state, in part, —"99.9 percent by number of the particles under one micron account for over 90 percent of the projected area of staining power but only about 25 percent of the total volume".

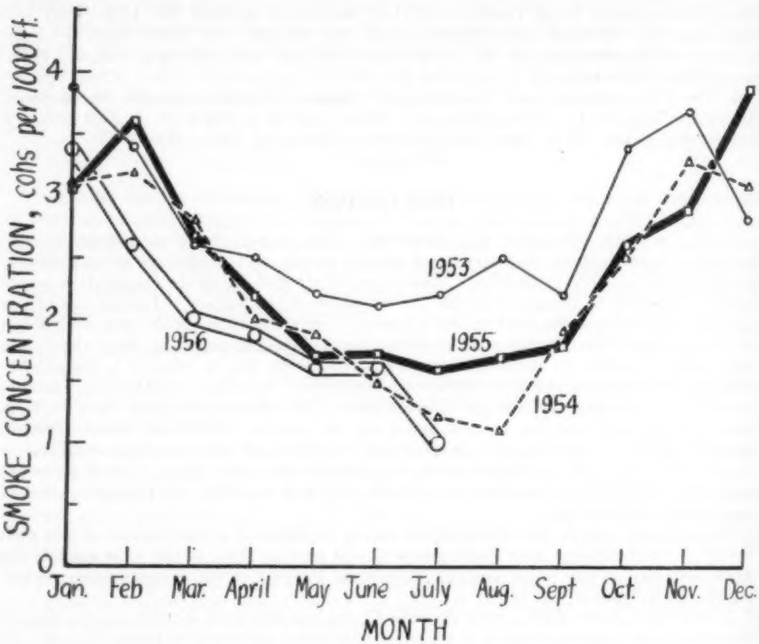


FIG. A—SMOKE CONCENTRATIONS (MONTHLY AVE.) FOR OAKLAND SECTION OF PITTSBURGH, 1953-1956

Let me repeat that Dr. Whitby and his associates have made important contributions to our knowledge of air-borne particulate. They are to be congratulated for these achievements and encouraged to carry the program still farther.

C. L. HEMEON, Pittsburgh, Pa. (WRITTEN): This paper is essentially an outline description of the techniques employed and results obtained in preliminary studies conducted to obtain a better understanding of the physical characteristics of air-borne dust incident to a major program of research related to the development of test methods for rating air cleaners. The content of this paper will have historical interest, we believe, because of the more interesting accomplishments being built on such foundations and which will doubtless be the subject of future papers by the authors.

A method is described in which rate of dustfall can be determined for time periods as short as one day, or less. It involves measuring the decrease in light transmission

through a glass slide that has been exposed at some location of interest. The authors propose a unit, *O.D.D.*, and indicate an empirical relationship between that unit and weight units, i.e., 1 *O.D.D.* = 1.66 tons per square mile per month.

We have been quite unable to follow this overly abbreviated discussion and because we believe it may be of interest to others, we should like to suggest to the authors that they extend their remarks in explanation of the interrelation between the units *O.D.*, *O.D.D.* and dustfall in weight units so that the Society Transactions will include a clear exposition of this subject.

The authors present results on weight concentration of suspended solids in various locations. The indicated average outdoor concentration is a little larger than 0.03 grains per 1000 cubic feet. They observed with regret that they could not maintain continuous sampling throughout a complete year's cycle.

Table A—Results of 12-Month Survey in Small Ohio Valley Town—Average Concentrations

Month	Concentration, Grams per 1000 Cu Ft	Month	Concenration, Grams per 1000 Cu Ft
February	0.03	June-July	0.02
March	0.05	August	0.02
April (1)	0.04	September	0.02
April (2)	0.03	October	0.03
May (1)	0.07	November	0.04
May (2)	0.03	Dec.-Jan.	0.04
May (3)	0.01	February	0.03
June	0.03		
		Weighted Ave.	0.032

We have some such data (Table A) from a survey we conducted spanning a 12-month period which may be of interest as a supplement to those presented in the present paper. The sampling device was a domestic type electronic precipitator with a capacity of 1200 cfm drawing on outdoor air. It was located in a small Ohio valley town in which soft coal was common domestic fuel. Sampling periods were consecutive but of irregular duration, ranging from 10 days to 6 weeks. The results given in Table A indicate average concentrations of the same order of magnitude as those reported in the present paper.

Similarly we can supplement the authors' data on staining concentrations over a 4-year period and according to month of the year. Unlike the monotonous regularity of the weight concentrations, one perceives in Fig. A a characteristic seasonal variation with a 4-fold range in monthly averages with a low of 1 and a high of 4 cohs per 1000 feet.

Authors' Closure (Mr. Whitby): The authors appreciate very much the help that Mr. Logsdon and his associates have given during this dust survey. I am sure that he is well acquainted with the problems encountered in making it.

These problems may be illustrated by citing one example. The small diaphragm pumps used for sampling occasionally developed valve trouble after which they sounded some thing like a sick cow. On two occasions this ended the sampling in commercial buildings.

The filter loading survey referred to by Mr. Logsdon consists of measuring the filter pressure drop during the life of 4 representative filters installed in special samplers,

operated in a variety of locations. Thorough analysis of the dust removed from the filters is providing much more information on the amount of fibrous particles in air-borne dust. Results confirm the figures given in this paper.

There are no methods of particle size analysis available to us which do not destroy most of the soluble particles. The centrifuge sedimentation analysis destroys all acetone soluble material as does the electron microscope technique. Light microscope techniques require the use of immersion oil which will dissolve many particles. The techniques for evaluating soluble or volatile particles are so elaborate as to make them impractical for this type of survey.

In regard to the agglomerating effect of the dissolved millipore filter mentioned by Mr. Logsdon, I believe that we have satisfied ourselves fairly well that the millipore in solution has no adverse effects. About 98 percent of the dissolved millipore is removed by syphoning of the supernatent liquid before the actual sedimentation.

In general we agree with Mr. Logsdon's statements regarding the effects of variations in air-borne dust on air cleaner evaluation. Future papers will detail our studies of this subject.

At Mr. Hemeon's request we are pleased to include the following additional remarks about the dust fall unit *O.D.D.*

By definition 1 *O.D.D.* is the amount of dust settled on a slide per day that will result in 0.001 optical density. Therefore

$$O.D.D. = \frac{10^3}{t} \times O.D. \qquad (2)$$

where

$$t = \text{days}$$

$$O.D. = \log_{10} \frac{I_0}{I}$$

From empirical measurement Engebretson found for dust fall on such slides that:

$$1\ O.D. = 1.88 \text{ mg per sq cm} \qquad (3)$$

Equating 2 and 3 it is found that:

$$1\ O.D.D. = 1.88 \times 10^{-3} \text{ mg per (sq cm)(day)}$$

$$= 1.66 \text{ tons per square mile-day} \qquad (4)$$

We are pleased that the dust concentration data given by Mr. Hemeon confirms the findings of this paper. Our values confirm the results obtained by the U.S. Public Health Service in their dust survey work.

1628

No. 1628

NATURAL CONVECTION COOLING AND DEHUMIDIFYING

By L. G. SEIGEL*, ERIE, PA., AND W. L. BRYANT†, CLEVELAND, OHIO

BASIC DATA for the design of natural convection cooling equipment which may be required for certain applications have not been available. Therefore, this investigation was undertaken to establish such data so that the performance of gravity type cooling and dehumidifying equipment might be predicted.

In order to accomplish this purpose, data were obtained from tests of both bare and finned single tubes and from tests of full size *gravity coil units*. All tests were run in the range of comfort air conditioning, and both sensible cooling and dehumidification were studied.

DESCRIPTION OF EQUIPMENT TESTED

The cooling equipment tested consisted of the following items:

1. One horizontal bare tube.
2. One horizontal finned tube.
3. One complete gravity cooling coil.

The bare tube was a ½-in. nominal size copper tube, 66.5 in. long. The actual outside diameter was 0.686 in. and the inside diameter was 0.605 in.

The finned tube was a copper tube, 67.5 in. long, and 0.638 in. outside diameter. The inside diameter was 0.590 in. The tube was fitted with aluminum fins spaced 3.12 per inch. The fins were 0.0095 in. thick and 2.0 in. square. Fig. 1, is a photograph of a section of this tube. After completion of one set of tests for this tube, the fin spacing was doubled by removing every other fin, and the tests were repeated. Thus, two fin spacings were tested. The gravity coil was a 5G Standard Navy Gravity Cooling Coil. It had a surface area of 128 sq ft obtained by using

---

*Director, Engineering Department, Gannon College. Associate Member of ASHAE.
†Associate Professor of Heat Power, Case Institute of Technology. Member of ASHAE.
Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, Pa., January 1958.

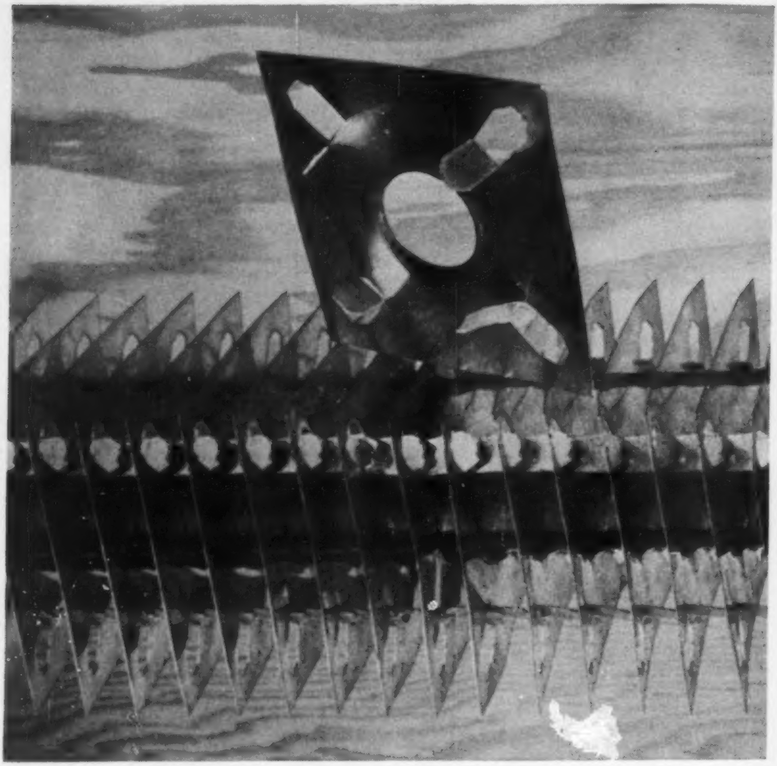


FIG. 1—PHOTOGRAPH OF FINNED TUBE USED IN TESTS

0.020 in. thick aluminum fins spaced 3 per in. on two horizontal rows of ⅝-in. copper tubes.

TEST METHOD AND PROCEDURE

The gravity coil was installed in a thoroughly instrumented metal-lined test room as shown in Fig. 2. Heat and moisture were supplied to the room by electric heaters and evaporators. Room dry-bulb temperatures were determined by means of vertically spaced thermometers and thermocouples as shown in the figure. For measuring relative humidity, 2 aspirating psychrometers were used. One was equipped with a sampling tube to measure the average condition in the space and the other was set to measure local humidity near the observation window.

Water temperatures were measured by mercury thermometers installed at the water inlet and outlet of the coil. This temperature difference across the coil was

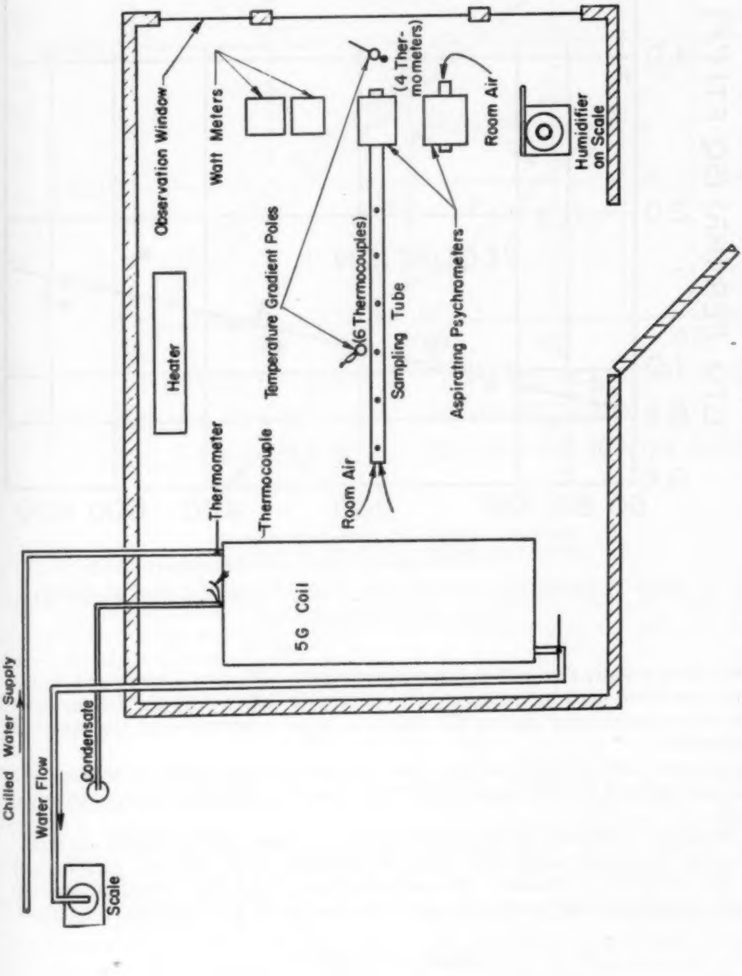
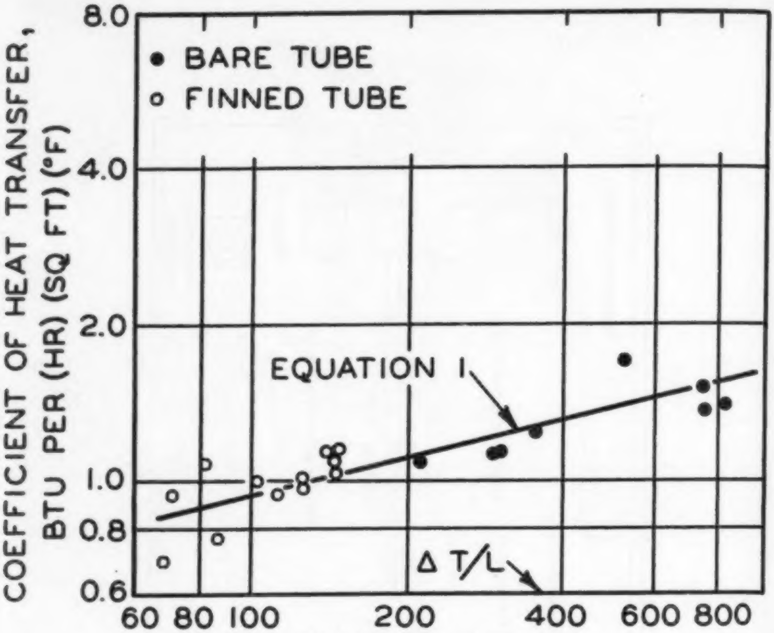


FIG. 2—SCHEMATIC DIAGRAM OF SETUP FOR TESTING GRAVITY COILS



Bare Tube: 0.686-in. OD, 0.605-in. ID copper tube
Finned Tube: 0.638-in. OD, 0.590-in. ID copper tube
3.12 fins per inch; 2- x 2- x 0.0095-in. thick aluminum

FIG. 3—HEAT TRANSFER COEFFICIENTS FOR GRAVITY COOLING WITH HORIZONTAL BARE AND FINNED TUBES

used with the water flow rate to determine the total coil load. Latent load for the space was determined by measuring the condensate collected by the drip pan and this value was checked against the weight of water evaporated by the electrical evaporators.

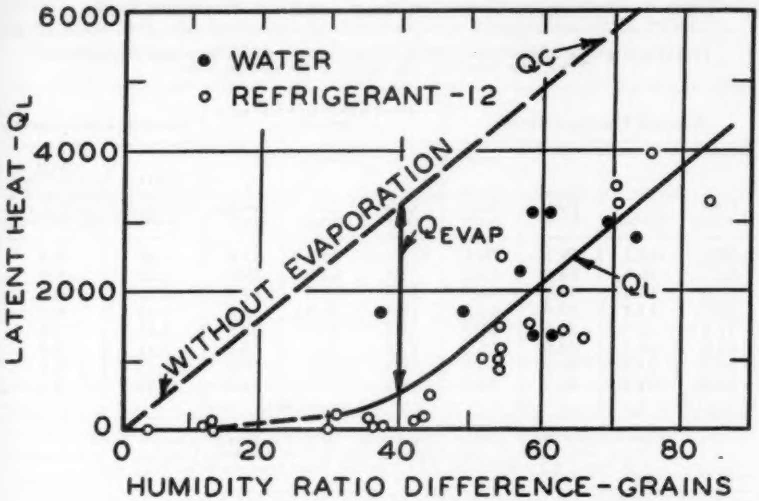
The space sensible heat load was then computed as the difference between the coil heat load and the heat equivalent of the collected condensate removed from the room.

The single tubes were tested in the same test space and by similar methods. However, during the single tube tests, an *equipment access wall* was used instead of the observation window. All instrumentation and flow connections were made through this wall so that the need for entering the test room was minimized.

RESULTS OF TESTS

For the single tubes, the results of heat transfer tests were found to agree with the following equation for both bare and finned tubes:

$$h_o = 0.29\,[(t_a - t_s)/L]^{1/4} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$



Q_C = LATENT HEAT REMOVED BY COIL
Q_L = LATENT HEAT REMOVED FROM ROOM

FIG. 4—TEST RESULTS FOR GRAVITY COIL

A plot of the test results and of this equation is shown in Fig. 3.

Results of tests when mass transfer was involved were not so consistent as those for heat transfer due to difficulties encountered in measuring the small condensate quantities and due to re-evaporation of the condensate from the collection pan. These results are shown in Table 1.

The mass transfer may be determined from the following equation using values of K_o from Table 1.

$$M_v = K_o A_o (w_a - w_s) \quad \text{. . . . .} \quad (2)$$

For the full size gravity coil, the test results are shown in Fig. 4. The points show considerable scatter, probably due to the difficulties encountered in making accurate measurements of very small heat and moisture quantities. However, a definite trend was established, and reasonable agreement is demonstrated with data obtained previously from tests of a similar coil using Refrigerant-12 as the refrigerant instead of water.

In the interpretation of the results for the gravity coil, it will be helpful to present the factors affecting the performance of natural convection cooling equipment.

When a gravity cooling coil is installed in a space both the heat removed by the coil and the mass or condensed water removed from the space must be considered. It was experimentally observed that all the vapor converted to water at the coil surface does not appear as removed condensate since some evaporates from the drip pan and some is by-passed in the form of fog or moisture particles.

TABLE 1—RESULTS OF COMBINED HEAT AND MASS TRANSFER TESTS

PERFORMANCE OF A SINGLE BARE COPPER TUBE COOLING AIR BY NATURAL CONVECTION
TUBE SIZE: 0.686 IN. OD, 0.605 IN. ID, 66.5 IN. LONG

Average Temperatures, F				Heat Removed by Tube Btu/hr			Transfer Coefficients	
Tube Surface	Water	Room Dry-Bulb	Room Dew-Point	Radiation	Convection	Condensation	Heat, Btu per (hr) (sq ft) (F deg)	Mass, lb H_2O per (hr) (sq ft) (lb H_2O per lb dry air)
52.5	48.2	89.3	70.1	19.1	53.4	45.6	1.45	5.8
52.1	46.8	94.8	73.4	22.5	64.0	89.4	1.50	8.9
51.6	46.4	95.9	73.2	23.3	67.3	87.9	1.52	8.8
46.7	42.1	83.9	63.5	18.5	53.9	37.6	1.45	6.2
47.1	42.3	84.5	61.9	18.8	54.3	28.5	1.45	5.6
46.0	43.2	82.0	51.3	17.9	51.8	5.5	1.44	3.6
46.3	43.8	80.2	53.7	16.7	48.5	9.9	1.43	4.5
48.0	44.8	84.5	59.3	18.4	52.5	31.9	1.44	8.2

PERFORMANCE OF A SINGLE HORIZONTAL ALUMINUM-FINNED COPPER TUBE
TUBE SIZE: 0.638 IN. OD, 0.590 IN. ID, 67.5 IN EXPOSED LENGTH;
FIN SIZE: 2 x 2 x 0.0095 IN.; 3.12 FINS PER INCH; SURFACE AREA: 11.75 SQ FT

69.5	49.7	87.1	78.0	63.7	208	225	1.0	3.5
67.4	44.8	87.4	81.6	71.1	256	210	1.1	2.0
69.7	58.5	87.4	79.3	62.9	212	296	1.0	4.1
68.0	54.3	89.6	84.8	76.6	266	372	1.1	2.6
73.0	54.2	89.7	84.2	59.2	205	222	1.0	2.3

PERFORMANCE OF A SINGLE HORIZONTAL ALUMINUM-FINNED COPPER TUBE
TUBE SIZE: 0.638 IN. OD, 0.590 IN. ID, 67.5 IN. EXPOSED LENGTH;
FIN SIZE: 2 x 2 x 0.0095 IN.; 1.56 FINS PER INCH

59.2	45.0	86.4	84.5	99	179	499	1.04	5.0
65.2	53.3	88.0	88.0	81	143	483	0.99	4.5
68.3	58.5	87.7	85.0	71	117	402	0.95	5.3
77.0	66.1	91.6	91.3	55	82	383	0.89	4.7
75.9	64.0	92.9	92.2	64	99	489	0.92	5.3

Note: The effect of fin spacing on mass transfer is shown by the last part of the above table. Increasing the fin spacing decreased the moisture concentration between fins and caused an increase in coefficient.

The presence of an unknown amount of fog and the re-evaporation of moisture from the coil surface and drip pan complicates the problem to the extent that the actual states of the air entering and leaving the coil are not known. Therefore, it is not possible to evaluate coil performance in the usual way. In order to overcome this difficulty and to permit coil computations to be based on average space air conditions, it is convenient to consider the coil to be in series with an adiabatic evaporator to take care of the by-passed fog and moisture. Through the operation of this imaginary evaporator, the average space air condition is converted into an unknown state at the coil.

Based on these considerations, the heat and mass transfer rates are specified by unit heat and mass transfer factors for the coil rather than by the familiar coeffi-

cients. The process of fog recirculation from coil exit to entrance can be accounted for by these unit transfer factors as indicated by the equations which follow.

The sensible heat removed from the space may be expressed as:

$$Q_s = (hA)_u(t_a - t_s) \quad \ldots\ldots\ldots\ldots \quad (3\text{-a})$$

or

$$Q_s = h_oA_o(t_1 - t_s) + CM_v h_{fg} \quad \ldots\ldots\ldots\ldots \quad (3\text{-b})$$

The latent heat removed from the space depends directly on the mass transfer relations as follows:

$$Q_L = (1 - C)M_v h_{fg} \quad \ldots\ldots\ldots\ldots \quad (4\text{-a})$$

$$Q_L = (KA)_u(w_a - w_s)h_{fg} \quad \ldots\ldots\ldots\ldots \quad (4\text{-b})$$

$$Q_L = K_oA_o(w_1 - w_s) - CM_v h_{fg} \quad \ldots\ldots\ldots\ldots \quad (4\text{-c})$$

If the re-evaporation or fog by-pass is reduced to zero, Equations 3-b and 4-c become the familiar heat and mass transfer equations, and computations may be made in the usual manner. However, for gravity coils, the by-passed moisture is not an insignificant value, and computations must be based on unit factors through equations of the form given in Equations 3-a and 4-b.

In the range of $(t_a - t_s)$ tested (about 30 to 40 F) the heat transfer factor divided by the coil surface area has been found to be close to unity throughout the range. However, the mass transfer factor becomes substantially zero (as indicated in Fig. 4) at a humidity ratio difference of approximately 35 grains per pound of dry air.

The practical significance of re-evaporation as brought out in Fig. 4, is that for space latent heat loads of zero or near zero the specific humidity difference $(w_a - w_s)$ will be approximately 35 grains. This potential difference is set up by fog by-pass and the moisture which is re-evaporated. If this were not the case, the dew-point in the space would approach the temperature of the coil surface when the space latent load is reduced to zero.

DISCUSSION OF RESULTS

The results of these tests indicate that, for constant sensible and latent loads, the minimum absolute humidity that can be maintained in a space cooled by the gravity cooling coil tested is about 35 grains per pound of dry air higher than the humidity corresponding to the refrigerant temperature. This means that the minimum space dew-point that can be maintained by cooling with gravity coils will be about 60 F when using chilled water at 45 F as refrigerant.

However, if the latent load is not constant and is reduced to zero as would be the case in an airtight compartment after closing it for storage, the difference in absolute humidity depends on the temperature difference between dry-bulb and refrigerant. It has been experimentally determined from unsteady-state tests conducted during this investigation, that the *humidity pull-down* or residual absolute humidity that can be maintained in a space having a zero continuous latent load will be 0.75 grains per pound of dry air higher than the humidity corresponding to the refrigerant temperature, for each Fahrenheit degree difference in temperature between the space dry-bulb and the refrigerant. For example, a space fitted with this gravity coil operating at 50 F refrigerant and a dry-bulb of 80 F would have a minimum absolute humidity of 0.75 (80 − 50) = 22.5 grains per pound of dry air higher than the hu-

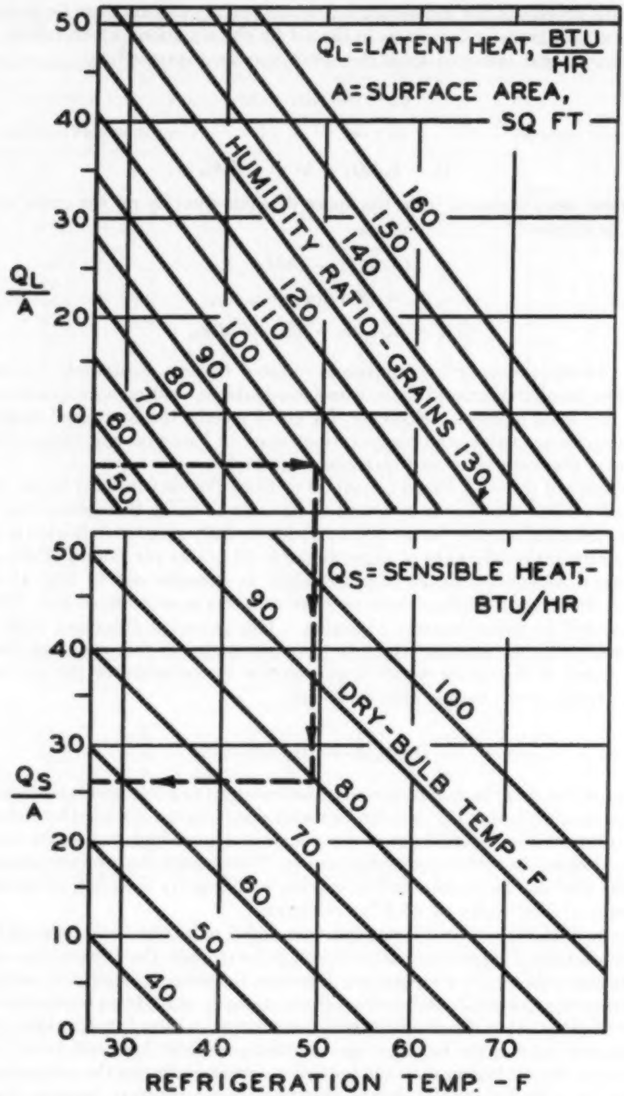


FIG. 5—PERFORMANCE CHART FOR GRAVITY COOLING COIL. T_R IS THE INLET WATER OR REFRIGERANT-12 SUCTION TEMPERATURE IN FAHRENHEIT DEGREES

midity corresponding to 50 F. This is 54 + 22.5 = 76.5 grains per pound of dry air. This relation is valid until approximately 35 grains difference is reached. For differences greater than 35 grains, it is no longer possible to maintain zero latent load and the steady-state performance of Fig. 4 applies.

APPLICATION OF RESULTS OF FULL SIZE COIL TESTS

To simplify the application of the results obtained from this investigation, a performance chart similar to that shown in Fig. 5 may be drawn for any particular coil design. This chart may then be used for all applications where there are continuous latent and sensible loads to be handled. If the latent load should be reduced to zero as might be the case when storing dry products for prolonged periods, the space conditions which will be maintained depend upon the re-evaporation characteristics of the space and the coil as described under Discussion of Results.

To illustrate the use of the coil data presented in this paper, the following examples have been prepared:

Example 1: *A space used for storage is to be maintained at* 76 *dry-bulb and* 50 *percent relative humidity. The sensible heat gain for the space is* 8400 *Btu per hr, and there is no ventilation or latent heat load during storage. Before closing, the space was ventilated with air at* 96 *dry-bulb and* 75 *wet-bulb. Chilled water is available at* 50 *F for cooling. Select a gravity coil to maintain the space at the storage conditions.*

Solution: 1. From Fig. 5, the coil capacity for 50 F water is found to be 22 Btu per (hr) (sq ft) at 76 dry-bulb.

2. To carry a load of 8400 Btu per hr then 8400/22 = 384 sq ft of coil surface are required.

3. The absolute humidity in the space will be determined by the difference between the space dry-bulb and the refrigerant temperature. For the temperature difference (76 — 50), the space absolute humidity will be 26 × 0.75 = 19.5 grains per pound of dry air above the humidity corresponding to 50 F refrigerant temperature. Therefore, the space absolute humidity will be 54 + 19.5 = 73.5 grains per pound of dry air.

4. At 76 dry-bulb and 73.5 grains per pound of dry air, the relative humidity is 56 percent, which is higher than the 50 percent required. If the 56 percent humidity is not satisfactory, the water temperature must be reduced, or the condition cannot be met by use of gravity coils of this design using a water temperature of 50 F.

5. If the water temperature is reduced to 45 F, the absolute humidity would be 0.75 (76 — 45) + 44 = 67 grains per pound of dry air, and the relative humidity at 76 dry-bulb would be 50 percent. Therefore, the condition could be met with 45 F water instead of 50 F.

For automatic control under light load conditions, reheat should be used regulated by dry-bulb thermostat. Control by cycling the coil is not recommended because it would cause high humidity during the *off* periods.

Example 2: *Assume that the space in Example* 1 *has a continuous latent load of* 2000 *Btu per hr and that the sensible load is* 9500 *Btu per hr. Let the required space condition be* 85 *dry-bulb and* 65 *percent RH* (*relative humidity*) (118 *grains*). *Chilled water is available at* 50 *F. Select a gravity coil to maintain the space conditions and prescribe a method of control.*

Solution: 1. From Fig. 5, the sensible load is 31 Btu per hr (sq ft) at 85 dry-bulb and 50 F refrigerant and the latent load is 15.6 Btu per (hr) (sq ft) at 118 grains and 50 F refrigerant.

2. For the sensible load, 9500/31, or 307 sq ft of coil surface are required, and 2000/15.6, or 128 sq ft are needed for the latent load. A standard coil size of 335 sq ft may be selected.

3. If the latent load is constant at 2000 Btu per hr, the coil latent load per square foot will be 2000/335 = 6.0. According to Fig. 5, this loading can be carried with 50 F refrigerant temperature when the space humidity is 103 grains. Therefore, the space will come to equilibrium at 58 percent RH when the dry-bulb temperature is 85 F. If less than 65 percent RH is not satisfactory, some type of humidification will have to be added to raise the space humidity since the coil area in this case is determined by the sensible load.

4. If it is desired to maintain exact space conditions at all times, elaborate control methods are required to provide both reheat and re-humidification. However, if it is necessary only to maintain humidity *below* a specified value, the control system may be simplified. To meet these requirements, it should be possible to have a space humidistat regulate the chilled water valve in parallel with a dry-bulb thermostat. For conditions where the sensible load is light and the latent load is high, the humidistat would call for more water than required to maintain the dry-bulb and reheat would be required (from an auxiliary thermostat or a low limit position of the dry-bulb thermostat regulating the water). For high sensible loads with light latent loads, the dry-bulb thermostat would control the water flow and the humidity in the space would be lower than the setting of the humidistat.

APPLICATION OF RESULTS OF SINGLE TUBE TESTS

In the case of the single tubes tested, fog by-pass was not evident within the range of refrigerant temperatures used. Re-evaporation relative to the space volume was small so that the room condition was considered equivalent to the condition of the air passing over the tube. Under these conditions it was possible to determine the re-evaporation by actual measurement and Equation 3-b could be used. Then, by substituting the mass transfer equivalent for the re-evaporation, Equation 3-b may be written as:

$$Q_s = h_t A_e(t_a - t_s) + CK_e A_e(w_a - w_s)h_{fg} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (5)$$

Note that h_t has been substituted for h_c to include the radiation as well as the convection heat transfer. Thus, Equation 5 gives the total sensible heat transferred from the space by the tube-drip pan combination.

The latent heat transfer for single tubes may be expressed by the following equation:

$$Q_L = (1 - C)K_e A_e \times (w_a - w_s)h_{fg} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (6)$$

The mass fraction of condensed water re-evaporated for the single tubes has been designated in Equations 5 and 6 by C. This factor, which depends on the rate of condensation and the design of drip pan, had approximate values of 0.5 for the bare tube and 0.1 for the finned tube over the range covered in these tests.

Use of the foregoing equations for single tubes will be demonstrated by an example which follows:

Example 3: Computations for a ⅝-in. OD Bare Tube. Assume conditions as follows: Room Ambient; 85 *dry-bulb,* 70 *wet-bulb,* 63 *dew-point. Load: Total heat* = 5500 *Btu per hr; Sensible heat* = 4400 *Btu per hr; Latent heat* = 1100 *Btu per hr; Load ratio* = 0.8. *Refrigerant: Chilled water at mean temperature of* 45 *F.*

Procedure for Solution: 1. The mean surface temperature of the tube, for practical purposes, may be considered equal to the mean water temperature, 45 F. This is accurate to about 2 percent for a bare tube.

2. It is next necessary to account for the sensible heat transfer by radiation. This may be done by use of a radiation coefficient which is added to the convection heat transfer coefficient.

The radiation coefficient is calculated by the equation:

$$h_r = [\sigma\epsilon(T_s^4 - T_a^4)]/(T_s - T_a) \quad \ldots\ldots\ldots\ldots \quad (7)$$

For this example,

$$h_r = 0.9 \text{ Btu per (hr) (sq ft) (Fahrenheit degree).}$$

3. The coefficient for convection heat transfer is calculated from Equation 1 or taken from Fig. 3.

$$h_c = 1.5 \text{ Btu per (hr) (sq ft) (Fahrenheit degree).}$$

4. The combined coefficient for convection heat transfer plus radiation is then:

$$h_t = h_r + h_c = 2.4 \text{ Btu per (hr) (sq ft) (Fahrenheit degree).}$$

5. The coefficient of mass transfer, K_c, is obtained from Table 1 for the bare tube.

$$K_c = 6.5 \text{ lb } H_2O \text{ per (hr) (sq ft) (lb } H_2O \text{ per lb dry air)}$$

6. The required area of the bare tube can now be calculated by using Equation 5. [$(w_a - w_s)$ = 0.006, from air-vapor tables.]

$$4400 = (2.4)\,(A_c)\,(40) + (0.5)\,(6.5)\,(A_c)\,(0.006)\,(1060)$$

$$A_c = 37.7 \text{ sq ft surface area}$$

7. The heat and mass transferred are then:

$$Q_s \text{ (heat by radiation, convection, plus evaporation)} = 4400 \text{ Btu per hr.}$$

$$Q_L = (1 - C)\, K_c A_c\, (w_a - w_s)\, h_{fg} = (1 - 0.5)\ 6.5 \times 37.7 \times 0.006 \times 1060$$

$$= 779 \text{ Btu per hr.}$$

Total heat transfer = 4400 + 779 = 5179 Btu per hr.

load ratio = 4400/5179 = 0.85

From these results it is seen that the required conditions cannot be met unless reheat is used (assuming that the water temperature cannot be reduced).

8. To meet the required condition, the tube area should be determined on the basis of the latent heat load. The required area is then:

$$A_c = 1100/(0.5)\,(6.5)\,(0.006)\,(1060) = 53.2 \text{ sq ft}$$

9. The heat and mass transferred are then: Sensible heat = 6207 Btu per hr; Reheat added to space = 1807 Btu per hr; Net sensible heat = 4400 Btu per hr; Latent heat = 1100 Btu per hr. Total heat removed by coil = 7307 Btu per hr; the Space load ratio = 4400/5500 = 0.8. Therefore, 325 ft of ⅝ in. OD tubing is required, representing 53.2 sq ft of surface area.

ACKNOWLEDGMENT

The authors gratefully acknowledge financial assistance from the United States Department of the Navy, Bureau of Ships, and their permission to publish the results of this study.

Credit is also due to Alfred Koestel and John B. Austin, Jr., of Case Institute of Technology, for assistance and suggestions in connection with the project.

REFERENCES

1. *Principles of Chemical Engineering*, by W. H. Walker, W. K. Lewis, W. H. McAdams and E. R. Gilliland (Third Edition, McGraw-Hill Book Co., Inc., New York, 1937).

2. *Introduction to The Transfer of Heat and Mass*, by E. R. G. Eckert (First Edition, McGraw-Hill Book Co., Inc., New York, 1950).

3. Deviation of the Actual Wet-Bulb Temperature from the Temperature of Adiabatic Saturation, by David Dropkin (*Engineering Experiment Station Bulletin* No. 23, Cornell University, July 1936).

4. Physics, by J. H. Arnold (Vol. 4, July 1933, pp. 255-262; Vol. 4, September 1933, pp. 334-340).

5. *Heat Transmission*, by W. H. McAdams (Second Edition, McGraw-Hill Book Co., Inc., New York, 1942).

6. A Precision Method for the Measurement of Condenser Tube Surface Temperatures for the Determination of Film Coefficients of Heat Transmission, by J. O. Jeffrey (*Engineering Experiment Station Bulletin* No. 21, Cornell University, 1936).

NOMENCLATURE

A = surface area, square feet.
A_c = heat or mass transfer surface area of coil or tube, square feet.
C = mass fraction of condensed water re-evaporated.
$\triangle W$ = humidity ratio difference, pounds of water per pound of dry air.
h_c = convection heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
h_{fg} = latent heat of vaporization of water (1060) Btu per pound.
h_r = coefficient of radiant heat transfer, Btu per (hour) (square foot) (Fahrenheit degree).
h_t = sum of convection and radiant coefficient, Btu (hour) (square foot) (Fahrenheit degree).
$(hA)_u$ = unit heat transfer factor, Btu per (hour) (Fahrenheit degree).
K_c = mass transfer coefficient, pounds per (hour) (square foot) $\triangle W$.
$(KA)_u$ = unit mass transfer factor, pounds of water per hour $\triangle W$.
L = height of heat path, feet*.
M_v = coil or tube mass transfer, pounds per hour.
Q_c = latent heat removed by coil.
Q_L = latent heat removed from the space, Btu per hour.
Q_s = sensible heat removed from the space, Btu per hour.
t_a = space dry-bulb temperature, Fahrenheit.
t_s = coil or tube surface temperature, Fahrenheit.
t_1 = dry-bulb temperature of air entering coil, Fahrenheit.
T_a = space dry-bulb temperature, Fahrenheit absolute.
T_s = coil or tube surface temperature, Fahrenheit absolute.
w_a = space humidity ratio, pounds of water per pound of dry air.
w_1 = humidity ratio of air entering coil, pounds of water per pound of dry air.
w_s = humidity ratio of saturated air at coil or tube surface temperature, pounds of water per pound of dry air.
σ = 0.171 = radiation constant.
ϵ = emissivity factor.

* This is the outside tube diameter for bare tube and fin height for finned tube.

1629

No. 1629

STABLE HOT-WIRE ANEMOMETER FOR LOW SPEEDS

By J. F. Kemp*, Pretoria, South Africa

IN LOW SPEED aerodynamics, and particularly in ventilation work, instruments are often required for measuring speeds ranging from a fraction of 1 fps to 15 fps and higher values. None of the existing types of low speed anemometer function as absolute instruments, so that each individual anemometer has to be calibrated against a suitable standard instrument. At speeds above 15 fps an NPL type Pitot-static tube, used in conjunction with a sensitive manometer, is the most suitable standard available. At lower speeds, the differential pressure generated by the tube becomes too small to measure with convenience and accuracy, and resort has to be taken to other types of instrument.

Several proprietary instruments are available for measuring in the range below 15 fps but these invariably fail in one or more respects when the following requirements are stipulated: (1) effective operational range to cover speeds from 0.1 fps to 15 fps, (2) calibration to be permanent, and (3) effects of variations in ambient atmospheric conditions to be compensated for, either automatically in the operation of the instrument, or by virtue of corrections applied to speed measurements.

The second and third requirements are closely associated with the intrinsic accuracy of the instrument. A high intrinsic accuracy is really the basic requirement to be fulfilled if the instrument is to be employed as standard. Furthermore, by stipulating the forementioned range, provision is made for the availability of a *single* instrument to serve as standard at very low velocities. For much time is wasted, and calibrations become unnecessarily expensive, if 2 different standards have to be applied in covering a particular range.

The hot-wire anemometer described in this paper was therefore developed to fulfill the stipulated requirements and has been found to operate successfully in all respects.

Basic Considerations

If, initially, no other calibrated instrument is available, the calibration of a first standard is probably most accurately performed on a whirling arm according to

*Head of the Aeromechanics Division of the *National Mechanical Engineering Research Institute*, South African Council for Scientific and Industrial Research.

Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, Pa., January 1958.

the technique developed by Ower and Johansen[1]. The method involves the use of a full-size model A of the anemometer B. the calibration being carried out in 2 steps: B is first mounted on the arm and is moved at various constant ground speeds in an enclosed duct, the instrument indication being noted at each speed. A, the model, is then mounted on the arm, and B set up stationary inside the duct, with its velocity sensing element suitably positioned to measure its own swirl speed, so to speak, as the model is moved over the same range of ground speeds as before. Since swirl speeds are normally very much lower than the relevant ground speeds,

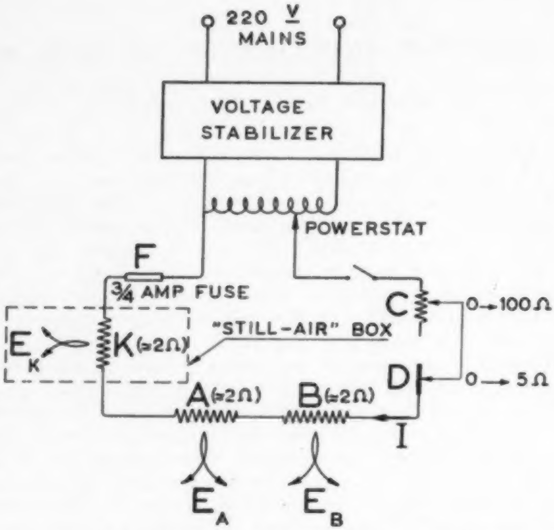


FIG. 1—HOT-WIRE SYSTEM EMPLOYING CONSTANT HEATING CURRENT

the appropriate relative air speeds can be obtained by a graphical process of iteration and the final calibration curve plotted accordingly.

Its high sensitivity at very low speeds, in addition to the fact that its velocity sensing element can be made sufficiently small in size for the measurement of its *own* swirl velocities on the working circle of the whirling arm apparatus, are the main factors which contribute to the eminent suitability of the hot-wire anemometer as a first standard.

L. F. G. Simmons was responsible for one of the most notable improvements in hot-wire anemometry during recent years when he developed a shielded hot-wire instrument[2] to counteract mainly the adverse effects that dust deposition had on

[1]Exponent numerals refer to References.

the stability of instruments employing thin, bare heater wires. Unfortunately the range of the Simmons instrument extends to only 5 fps in the upper limit; in order to increase the range to higher values, resort needs to be taken to an improved method of controlling the heating current[2] if the accuracy of the instrument is to be maintained.

DEVELOPMENT OF A HOT-WIRE SYSTEM

Initially, therefore, the development of the new anemometer was centered upon the problem of devising a system in which the heating current could be con-

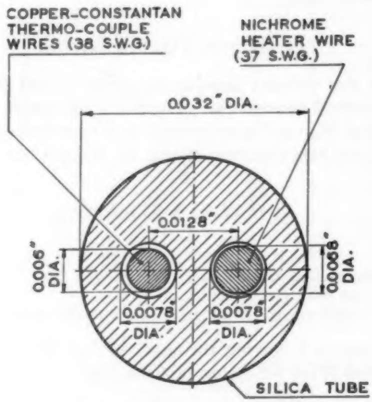


FIG. 2—DETAIL OF SIMMONS-TYPE ELEMENT

trolled within very fine limits. The velocity sensing element was to be shielded in the manner suggested by Simmons. The system finally adopted was essentially the same as that previously used by the author in connection with the measurement of axial components of velocity[3]. The modified circuit diagram is depicted in Fig. 1.

In this system, a constant current I heats a nichrome wire of about 1-in. length at B. The heater wire is threaded through one of the bores of a twin-bore silica tube, the other bore being used to accommodate the hot junction of a copper-constantan thermocouple, whose cold junction is exposed to the ambient temperature in the air stream. Thus element B, when placed in an airstream, serves as velocity sensing element, the emf, E_B, being an indication of air speed.

Current indication is effected by means of element K whose construction is identical to that of B. K, however is housed in a *still-air* box, where the air velocity is zero. Under specified ambient conditions, the emf, E_K, will be a function of heating current I only. Current control is effected by means of rheostat C (coarse control) and slide wire D (fine, stepless control). The voltage stabilizer further

enhances the accuracy of the control system, while fuse F safeguards the elements B and K.

The main advantage of the system is derived from the fact that current control is performed simultaneously with speed measurements. A second advantage is that an additional velocity sensing element, such as A, can be incorporated in the system. Thus, instead of using a model for calibration purposes in the manner already described, a second, working anemometer A can be constructed, and can be calibrated together with B with little additional effort.

A final advantage stems from the fact that the system lends itself readily to the application of the laws of heat transfer from slender cylinders in an airstream. Formulae can be derived to indicate the effects that variations in heating current and in ambient conditions have on the accuracy of speed measurement. These aspects will be considered next.

THEORETICAL CONSIDERATIONS

Heat transfer from the velocity sensing elements A and B (Fig. 1) takes place mainly by virtue of forced convection. Over a wide range of Reynolds number R_e, the Nusselt number N_u can be expressed as a function of R_e by the following relationship derived from experimental results by Hilpert[4] on slender heated cylinders in an airstream:

$$N_u = hd/k_w = 0.743R_e^{0.43} \quad \ldots \quad (1)$$

where

h = coefficient of heat transfer.
d = diameter of heated element.
k_w = thermal conductivity of air evaluated at the mean film temperature.

With an electric current I flowing through the hot-wire of resistance R, Equation 1 is applied to obtain the following relationship:

$$I^2R = c_1k_w(R_e)^{0.43}(T_w - T_a) \quad \ldots \quad (2)$$

where

c_1 = a constant.

$(T_w - T_a)$ denotes the effective temperature difference between the hot-wire element and the undisturbed airstream. This factor is measured in terms of an emf, (E, say), which may be assumed proportional to $(T_w - T_a)$ for the copper-constantan thermocouple over the operational range of temperatures of the element. Furthermore, the resistance R of the nichrome heater wire, whose temperature coefficient is extremely small, may be taken as constant.

Therefore,

$$I^2R = c_2k_w(v/\eta)^{0.43}E \quad \ldots \quad (3)$$

where

c_2 = a constant.
v and η = air velocity and kinematic viscosity respectively.

The current control element K loses heat by virtue of natural convection, and the Lorenz[5] equation for heat transfer may be applied in the form:

$$I^2R = c_3k_K\eta^{-0.5}(T_K - T_a)^{1.25}(T_a)^{-0.25} \quad \ldots \quad (4)$$

where

c_3 = a constant.
k_K = thermal conductivity of air at the mean film temperature of element K.
$(T_K - T_a)$ = effective temperature difference between the control element K and the airstream.

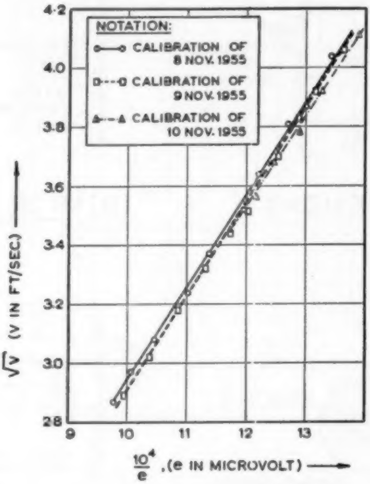


Fig. 3—Instability at Higher Speeds of Simmons-Type Element

In the present system $(T_K - T_a)$ is maintained at a constant value, which is proportional to E_K, the emf of the thermocouple of the control element. Division of Equations 3 and 4 yields the following result:

$$E = CE_K^{1.25}\eta^{-0.07}v^{-0.43}T_a^{-0.25} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (5)$$

C includes the ratio k_w/k_K, which varies only slightly with ambient atmospheric conditions, so that C may be taken as a constant.

Equation 5 presents the relationship between the various parameters involved in the operation of the hot-wire system, and will be used as basis for further analyses.

Accuracy of System Under Constant Ambient Conditions

If η and T_a in Equation 5 remain constant, it is easily shown, by partial differentiation, that

$$\Delta v/v = 2.33(\Delta E/E) + 2.90(\Delta E_K/E_K) \quad . \; . \; . \; . \; . \; . \; . \; . \quad (6)$$

In the system under consideration, E_K is measured by means of a field potentiometer having an accuracy of $\pm$ 2 microvolts, and is maintained at a value of

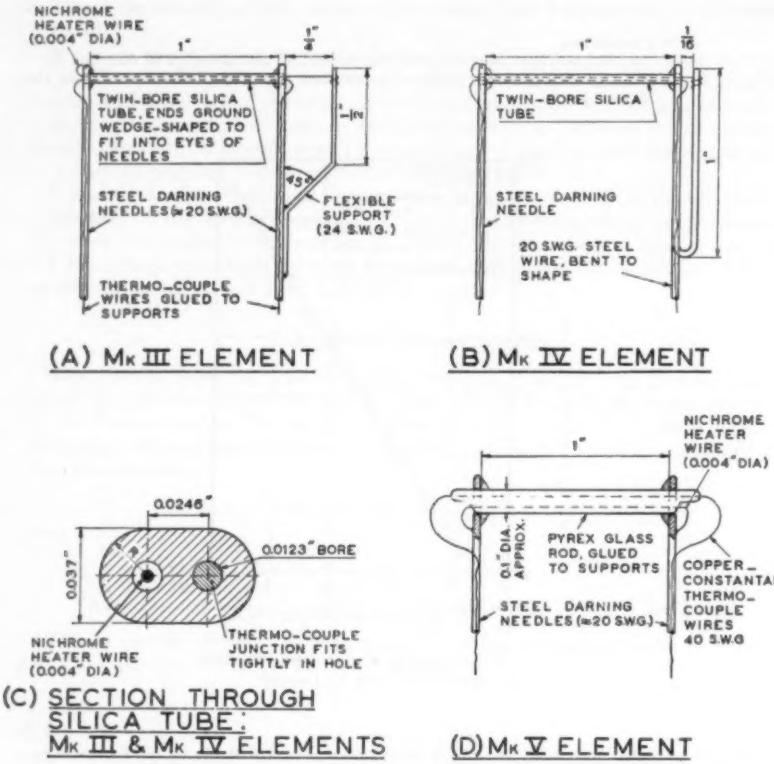


FIG. 4—CONSTRUCTIONAL DETAILS OF MK III, MK IV, AND MK V ELEMENTS

approximately 4000 microvolt. E is measured by means of a vernier potentiometer to an accuracy of $\pm$ 1 microvolts. At the upper limit of the speed range, *i.e.* at 15 fps E has its smallest value which is of the order of 750 microvolt. Under these conditions, the accuracy of the system is, theoretically, at its lowest and amounts to

$$(\Delta v/v \times 100) = (0.35 + 0.15) \text{ percent} = 0.5 \text{ percent}$$

Tests performed on 2 anemometers, which had been calibrated as a pair on the whirling arm, indicated that the agreement between the 2 instruments, when these were compared against each other at constant air speeds, closely followed the theoretical values calculated from Equation 5 down to a speed of about ½ fps where maximum accuracy was obtained. Below this speed, the agreement became less accurate, due rather to the method of calibration than any shortcoming in the operation of the anemometers, for it was found that weak thermal currents in the whirling arm duct began to have a detrimental effect on the calibration at very

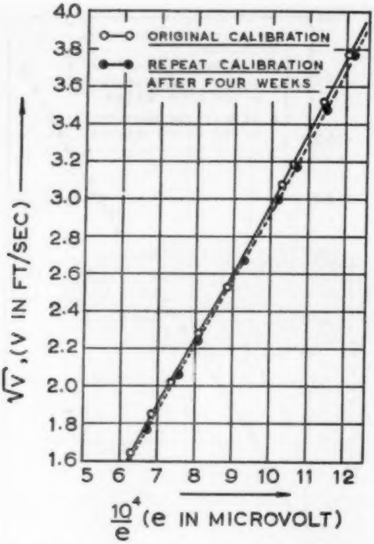


FIG. 5—INSTABILITY OF MK IV ELEMENTS

low speeds. At 0.1 fps which represents the lower limit of the range of the anemometers, the difference between the instruments amounted to 4 percent.

VARIATIONS IN AMBIENT CONDITIONS

If E_K in Equation 5 is maintained at a constant value, it is possible to deduce the effects on the accuracy of speed measurement of variations in atmospheric pressure, humidity and ambient air temperature. Furthermore, it can be shown that variations encountered from day to day in atmospheric pressure and humidity are of minor importance and that ambient air temperature is, in fact, the only variable that needs to be considered.

This was done in a previous paper[3] and the following expression was derived for temperatures T_a in the region of 20 C:

$$\Delta E/E = -0.38(\Delta T_a/T_a) \quad \ldots\ldots\ldots\ldots \quad (7)$$

where

ΔE = the correction to be applied to the emf reading E, when the temperature varies by an amount ΔT_a from the value T_a (in degrees centigrade absolute) which obtained during calibration.

With the aid of Equation 7 it also becomes possible to correct individual emf readings taken during the calibration of the anemometers on the whirling arm, and to prepare a calibration curve relating to a uniform air temperature which might

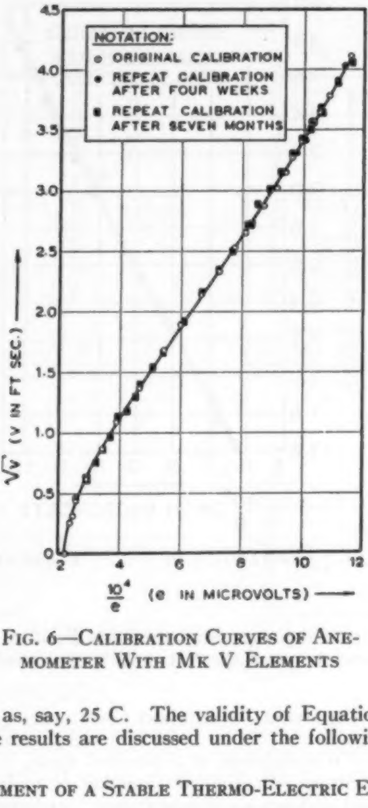


FIG. 6—CALIBRATION CURVES OF ANEMOMETER WITH MK V ELEMENTS

be chosen arbitrarily as, say, 25 C. The validity of Equation 7 was verified experimentally, and the results are discussed under the following section.

DEVELOPMENT OF A STABLE THERMO-ELECTRIC ELEMENT

The anemometers, and the still-air box containing the current control element, were initially equipped with shielded elements of the Simmons type in which a twin-bore silica tube, about 1 in. in length, was used to accommodate the heater wire and the thermocouple junction. A section through such an element is illustrated in Fig. 2. All the thermocouple junctions were welded, while the nichrome heater-wires were annealed by heating electrically and maintaining a red-hot temperature for several hours.

Altogether 4 anemometers and 2 still-air boxes equipped with these elements were thoroughly investigated and the tests repeatedly revealed instability of the elements, the phenomenon being most marked at higher values of air speed. Fig. 3, for example, illustrates how the calibration curves for a particular anemometer varied from day to day on 3 successive days.

The instability was eventually ascribed to the effect of thermal expansion and contraction of the heater wire in the silica tube. Since the 2 ends of the wire were welded to relatively sturdy steel supports, the wire was constrained in such a way

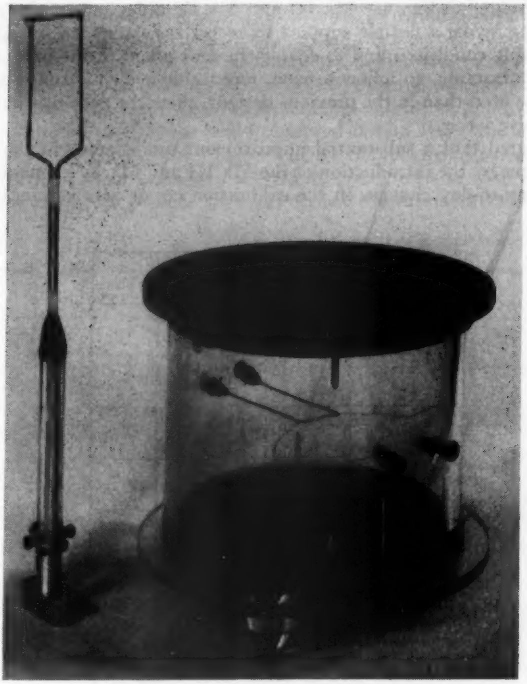


FIG. 7—PHOTOGRAPH OF HOT-WIRE ANEMOMETER AND STILL-AIR BOX

that expansion and contraction could only be taken up by a lateral movement of the wire. Considering the high temperature gradients present between the wire and the thermocouple junction, especially at high air speeds, it is evident that any lateral displacement of the heater-wire would affect the temperature at the thermocouple. Since the lateral movement of the wire would furthermore not follow the same cycle every time the heating current is switched on or off, sporadic changes might be expected to occur in the calibration of the instrument.

A second configuration of the thermo-electric elements was now constructed in which the welded thermocouple junction was enlarged to ensure a snug fit in the silica tube, while the heater wire was bent to form zig-zag kinks along its length, again with the motive of ensuring a snug fit. Although some improvement was effected by this construction, unpredictable changes in the calibration curve were again revealed by test, indicating that displacements of the wire relative to the thermocouple still persisted.

Three different designs, designated by Mk III, Mk IV, and Mk V, respectively, were next investigated on the whirling arm. Constructional details of these ele-

ments are presented in Fig. 4. Mk III and Mk IV differ only slightly from each other, each employing a flexible support to keep the heater-wire straight and taut under all heating conditions and to enable the movement of the wire, whether expanding or contracting, to follow a fixed, repeatable cycle. A more robust silica tube had to be used than in the previous designs, since the tube was now subjected to a compressive force.

Tests indicated that a substantial improvement was effected in the stability of the anemometer by the introduction of the Mk III and Mk IV elements, inasmuch as sporadic day-to-day changes in the calibration curve were eliminated. It was

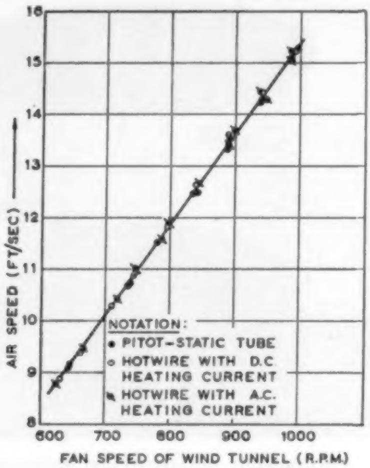


FIG. 8—COMPARISON BETWEEN ANEMOMETER AND NPL TYPE PITOT-STATIC TUBE

evident, however, that a slow but steady change did occur in the calibration. After using the anemometer with Mk IV elements daily for a period of one week after the original calibration, speed indications were found to be consistently lower, the difference at the maximum speed of 15 fps amounting to approximately ½ percent. After 4 weeks of continual use, the instrument was again calibrated, and it was found that the difference had increased to about 2 percent at 15 fps. The two calibration curves are illustrated in Fig. 5. The only possible explanations for this long-term instability of the elements were: (1) that the physical and electrical properties of the heater wire had changed as a result of an aging process, or (2) that dust particles had penetrated into the silica tube through the unsealed end on the side of the flexible support. The results that were subsequently obtained for the Mk V element proved, however, that no aging effects were present.

The Mk V element was finally tested and proved to possess the required long-term stability. In this element both heater and thermocouple wires are fused into

pyrex glass, the technique being briefly as follows: Two thin tubes of glass are first drawn and cut to the required length and are then fused separately on to the heater and thermocouple wires. Initially fusion is not carried out right up to the ends of the tubes since the thin wires easily burn when exposed to a flame. In the next step the two tubes are fused together and manipulated in the flame to form a solid rod near the middle, care again being observed not to bring the flame near the exposed wires at the ends of the tubes. Finally the exposed portions of the wires are threaded into 2 silica tubes of about ⅛-in. diameter, and with the wires thus protected against direct contact with the flame, the fusion process can be com-

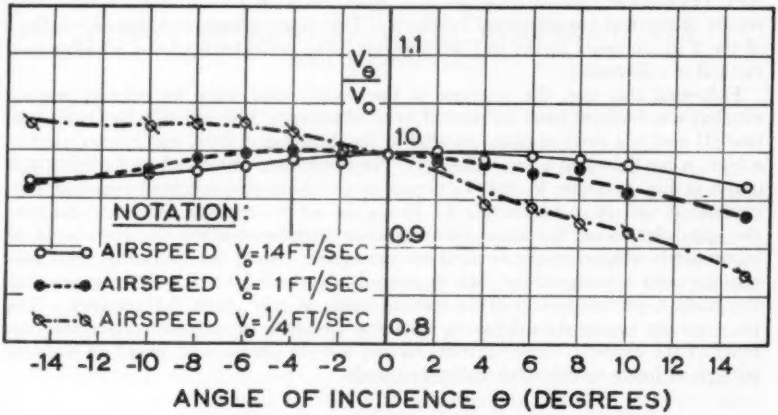


FIG. 9—DIRECTIONAL RESPONSE CURVES OF ANEMOMETER

pleted over the entire length of the element. In its final form the element therefore comprises the heater and thermocouple wires which are, in effect, fused into a solid pyrex rod. In this construction the wires are not only sealed against dust contamination, but are also prevented from moving relatively to each other or to the glass rod.

The calibration curve of the anemometer, equipped with Mk V elements, is depicted in Fig. 6. The solid points on the curve represent the results of a repeat calibration that was performed after the anemometer had been in daily use for four weeks. A third set of points denotes the results of yet a further calibration carried out after the instrument had been in service for seven months. All curves had been plotted for an arbitrarily selected uniform temperature of 25 C, by applying the formula given in Equation 7. The ambient air temperature had been very much the same for the first 2 calibrations, but was considerably lower during the third calibration when differences up to 11 C were recorded. If the third calibration curve is plotted without correcting the results to the reference temperature 25 C, differences of over 3 percent are encountered at higher speeds when this curve is compared with the original. By applying the temperature corrections, however, the curve is displaced to coincide with the original. Assuming that the

stability of the anemometer is proved by the results of the first 2 calibrations, the results of the third test can be taken as proof of the accuracy and usefulness of the temperature correction formula.

Fig. 7 shows the anemometer in its final form, together with its still-air box containing the current control element K (Fig. 1).

ADDITIONAL TESTS

Immediately after the original calibration had been completed, the hot-wire anemometer was removed to a 24-in. low speed windtunnel and the whirling-arm calibration checked in the speed range between 9 and 15 fps, using a calibrated NPL Pitot-static tube as standard in conjunction with a 13-in. manometer. The results of the test are presented in Fig. 8. The close agreement between readings of the 2 instruments serves to illustrate the basic correctness of the whirling-arm method of calibration.

Following this test, the response of the anemometer when its velocity sensing element was inclined from its normal vertical attitude (for which it had been calibrated) and in a vertical plane parallel to the direction of flow, was investigated at a high, a medium and a low velocity. The results are presented in the form of a graph in Fig. 9, where V_0 and V_θ denote respectively the indicated speed at zero inclination and at an inclination θ. From the set of curves, it is evident that, as the speed decreases, the errors that would be introduced if the element should be inaccurately aligned to the vertical are increased. At ¼ fps for instance the percentage error introduced for each degree misalignment up to $\theta = 10$ deg amounts to nearly 1 percent, while at 14 fps the value is only about 0.15 percent. The phenomenon obviously originates from the presence of natural convection currents at the element, these currents having a more pronounced effect on the flow pattern at lower speeds than at higher speeds.

ACKNOWLEDGMENTS

The work described in this paper was carried out in the *National Mechanical Engineering Research Institute* of the South African Council for Scientific and Industrial Research, as part of a thesis submitted to the University of Stellenbosch. The paper is published by permission of the University and of the South African Council for Scientific and Industrial Research.

The author also wishes to thank M. L. Strydom for his invaluable assistance during the development of this instrument.

REFERENCES

1. Ower and Johansen: R & M Report 1437 (1931).
2. Simmons: *Journal of Scientific Instruments*, Vol. 31 (1954).
3. Kemp: *Journal of the Institution of Heating and Ventilating Engineers*, Vol. 24 (December 1956).
4. Hilpert: *Forsch. Ing. Wes.*, Vol. 4 (1933).
5. Lorenz: *Ann d. Phys.*, Vol. 13 (1881).

1630

No. 1630

EFFECT OF HEATED-FLOOR TEMPERATURES ON COMFORT

By RALPH G. NEVINS* AND ARTHUR O. FLINNER,** MANHATTAN, KANS.

WITH INCREASED interest during the past 10 years in the use of floor-panel-heated residences and buildings, the effect of the floor surface temperatures on comfort has been the subject of discussion and research. At Kansas State College, beginning in 1950, a series of experiments were undertaken to determine the effect of floor surface temperatures on the sensation of comfort. This paper deals with the first phase of the program. It is concerned with the subjective reactions of college age students, both male and female, to various floor-panel-heated environments when the exposure period is 60 min and the students are seated at rest.

An exposure period of 60 min was selected for the experiments conducted during the first phase for 2 reasons: (*a*) the tests would provide comfort data for those applications where the period of occupancy was relatively short and of the order of 60 min, and (*b*) the period of 60 min made it possible to schedule the experiments during regular class periods which allowed a greater number of students to participate. It was also considered that these short time tests would be used as a guide in planning long time tests for the research program's second phase.

The second phase of the study will be undertaken during the winter of 1957-58 and will be concerned with the reactions of college age students to floor-panel-heated environments when the exposure period is 3 hr. The second phase program will include tests to evaluate the effect of floor temperatures on comfort for people seated at rest and standing doing light work.

Floor Temperatures In Practice: The design criteria in use today, in the United States, limit the floor surface temperature for floor-panel-heated installations to a maximum of 85 F[1]. Since the panel surface temperature determines the heat output of a given floor panel, a temperature of 85 F restricts the use of a floor-panel-heated system to those regions of the United States with moderate design temperatures. Even in these localities, buildings with large exposed glass areas require a supple-

---

*Professor and Head, Department of Mechanical Engineering, Kansas State College. Associate Member of ASHAE.
**Professor of Mechanical Engineering, Kansas State College, Member of ASHAE.
[1]Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, January 1958.

mental heating system. A survey of a floor-panel-heated residence in Manhattan, Kans., on a day when the outside temperature was 5 F showed floor surface temperatures of 88 F in the kitchen and 90 F in the bedrooms. This residence had an average amount of window area and had been constructed in about 1952.

The limiting temperature of 85 F seems to have been based on some experimental evidence but mostly it seems to have been an arbitrary figure. In 1949, Herrington and Lorenzi published a conclusion that floor temperatures above 75 F were not desirable[2]. This conclusion was based on the physiological consideration regarding the maintenance of a high vascular tone in the lower extremities.

Recent research in this field by F. A. Chrenko[3] in England indicates that floor surface temperatures should be limited to 75 to 77 F, especially if the occupants of the space are likely to be walking about for long periods. This conclusion was based on laboratory experiments using 5 men and 3 women with an exposure period of 60 min. Using a 10 percent incidence of discomfort the maximum desirable floor surface temperature for women sitting was 79 F and for men sitting was 84 F. For both men and women subjects, walking about, the maximum floor surface temperature was 80 F. Chrenko's results were in general agreement with present practice in England which is to limit floor surface temperatures to 75 F.

At the International Conference on Heating, Ventilating and Air Conditioning, held in Paris in 1955, F. C. Marechal presented a paper dealing with the permissible temperatures for ceilings and floors[4]. The experiments conducted in France by Marechal and more recently by Missenard were discussed. In Marechal's experiments using 2 subjects, the maximum foot temperature which the subjects could endure for long periods of time without discomfort was 87.8 to 88.7 F. Data were presented giving the air temperature as a function of floor temperature with the percentage of subjects expressing discomfort as a parameter, the latter also being correlated with the foot temperature. It is interesting to note that for a floor temperature of 75.2 F the air temperature which gives zero percent discontent is, from the American viewpoint, a chilly 63 F (by Marechal) or 68 F (by Missenard). For a floor temperature of 80.6 F the resulting air temperatures would be 60 or 64 F. It was indicated that comfort can be achieved by maintaining the temperature of the foot below 87 F and this can be accomplished by the proper combination of floor surface and air temperature.

Research at Kansas State College: To study the effect of floor surface temperatures on the comfort sensations of college age subjects, comfort data have been obtained using 108 male and 21 female college students. The male students ranged in age from 18 to 31 years with an average age of 22.9 years. The female students ranged in age from 19 to 22 years with an average of 19.7. The data were obtained by subjecting groups of approximately 20 to 25 students to various environments beginning in 1950. The data from each group were analyzed statistically.

EXPERIMENTAL ROUTINE

The subjects were exposed in a psychrometric room to each test environment for periods of 60 min. The subjects reported to the test room at least one hour after their last meal and directly from class or their lodging. Except for the 1954-55 tests, the subjects were not formally acclimated to some neutral environment before entering the psychrometric room. During the 1954-55 tests, the subjects were seated in an adjoining office for 30 min preceding the tests to allow them to

become somewhat acclimated to inside conditions. The subjects wore medium weight clothes and were tested 2 at a time.

The comfort sensations were obtained using a 7-point comfort scale shown as follows:

1. cold	4. comfortable	6. warm
2. cool	5. slightly warm	7. hot
3. slightly cool		

A thermocouple was placed on the ball of the subjects foot and for the 1953-55 tests an additional thermocouple was placed on the index finger.

Subjects were instructed prior to the tests regarding the use of the comfort scale and the test procedure. The subjects remained seated with their feet resting on the floor throughout the test period and either read or studied. The subjects were seated in chairs which were located in the same position for each test. The chairs were in thermal equilibrium with the surroundings for each test. Comfort votes and other data were recorded at 15-min intervals.

The psychrometric room is a room 10 ft on a side with a 10-ft ceiling located within a larger room. The temperature of the floor and of one wall can be controlled independently from the air or other surface temperatures. Air is introduced into the psychrometric room through ventilation panels located in the ceiling and can be heated or cooled. The air exhausts through slots located around the edge of the room. Previous measurements indicated that the air velocity was less than 40 fpm below the 8-ft level and at the 4-ft level was less than 25 fpm. All surfaces of the psychrometric room were painted a flat machine grey.

Measurement of Physical Data: The temperatures of the floor panels in the psychrometric room were measured by copper-constantan thermocouples taped to the center of the 5 floor panels nearest the outlet header and to the center of the 5 floor panels nearest the inlet header. The floor surface temperature was assumed to be the average of the 10 temperature readings.

The air temperature in the psychrometric room was measured by shielded thermocouples located 6 in. from the ceiling, 5 ft from the floor, and 6 in. from the floor. These 3 thermocouples were located on a vertical line above a point 4 ft from the west wall and on the east-west center line of the floor. For the determination of the mean radiant temperature, an 8-in. diam globe thermometer was placed at the center of the room 5 ft above the floor. Surface temperatures of the walls and ceiling were determined by sets of 5 copper-constantan thermocouples connected in parallel. The velocity of the psychrometric room air was measured at the center of the room 4 ft above the floor.

Test Conditions: The environment to which the subjects were exposed was determined by floor temperature and an air temperature. Air temperatures of 65, 70, 75, 80, and 85 F and floor surface temperatures of 65, 70, 80, 85, 90, 95, and 100 F were used. The temperatures of the room surfaces other than the floor could not be controlled and therefore varied somewhat with each condition. However, before data were taken, these surfaces were allowed to reach an equilibrium condition and the temperatures of these surfaces were usually within ± 3 F of the air temperature.

The dew-point in the psychrometric room could not be regulated and therefore the relative humidity to which the subjects were exposed varied from 30 to 60

TABLE 1—CORRELATION COEFFICIENTS FOR COMFORT VOTE WITH VARIOUS PHYSICAL PARAMETERS FOR COLLEGE AGE MEN SEATED AT REST, 60-MIN EXPOSURE 1950-51 AND 1954-55 TESTS

COMFORT VOTE	COEFFICIENT
With Air Temperature	$r = 0.91$
With Effective Temperature	$r = 0.9395$
With Operative Temperature	$r = 0.9272$
With Floor Temperature	$r = 0.0341$ [a]
With Floor Temp Air Temp = 75	$r = 0.184$
With Floor Temp Air Temp = 70	$r = 0.31$
With Floor Temp Air Temp = 80	$r = 0.460$
With Floor Temp Air Temp = 85	$r = 0.197$
With Relative Humidity	$r = 0.0462$ [b]
	$r = 0.0688$ [c]

[a] All air temperatures considered.
[b] Spring 1950 tests only.
[c] 1957 tests, air temperature 75 F.

TABLE 2—COMFORT VOTE AT THE END OF A 60-MINUTE EXPOSURE FOR MALE COLLEGE STUDENTS SEATED AT REST 1950-51 AND 1954-55 TESTS

AIR TEMP	FLOOR TEMP	NO. OF SUBJECTS	FINAL COMFORT VOTE	STANDARD DEVIATION
65	75	25	2.87	0.586
65	85	31	3.1	0.854
65	95	6	2.6	0.548
70	65	14	3.1	0.500
70	70	5	3.4	0.500
70	75	7	3.7	0.458
70	80	10	3.5	0.424
70	85	11	3.5	0.346
70	90	11	3.3	0.648
70	95	44	3.7	0.500
75	65	16	3.7	0.400
75	70	4	3.8	0.245
75	75	39	3.9	0.387
75	80	6	3.7	0.346
75	85	39	3.9	0.400
75	90	5	3.9	0.141
75	95	38	3.98	0.346
80	65	16	4.2	0.316
80	70	6	4.0	0.0447
80	75	7	4.6	0.519
80	80	7	4.2	0.173
80	85	5	4.2	0.436
80	90	2	4.0	0.000
80	95	3	4.3	0.300
85	65	16	4.98	0.872
85	70	2	4.88	1.020
85	75	33	5.22	0.854
85	80	4	4.42	0.663
85	85	36	5.13	0.775
85	90	3	4.57	0.490
85	95	34	5.56	0.755

percent. It has been shown that the effect of relative humidity on comfort† in the range of dry-bulb temperatures from 73 to 77 F is overemphasized by the ASHAE Comfort Chart[5,6]. Likewise the data obtained from these tests show that for an air temperature of 75 F and floor surface temperature of 75 or 80 F, the effect of relative humidity could not be detected throughout a range of relative humidities of 30 to 60 percent (See Table 1). On this basis the effects of the relative humidity on the comfort sensation were assumed negligible.

Results: The comfort votes recorded by college age males at the end of a 60-minute exposure to environments represented by the air temperature and the floor temperature are given in Table 2 for tests conducted during 1950-51 and 1954-55. The vote shown is the mean vote for the group of subjects exposed to a particular environment. As an index of the correlation of thermal sensation with parameters

TABLE 3—COMFORT VOTE AT THE END OF A 60-MIN EXPOSURE FOR COLLEGE AGE FEMALES, SEATED AT REST 1953-54 TESTS

AIR TEMP	FLOOR TEMP	NO. OF SUBJECTS	FINAL COMFORT VOTE	STANDARD DEVIATION
65	75	19	2.6	0.66
65	85	16	2.9	0.69
65	95	18	2.8	0.61
75	75	22	4.1	0.73
75	85	22	4.0	0.29
75	95	17	4.1	0.78
85	75	18	4.8	0.55
85	85	22	5.1	1.01
85	95	19	6.5	0.75

such as air temperature, floor temperature, effective temperature and operative temperature, the correlation coefficients were calculated and are tabulated in Table 1. It should be noted that these coefficients hold only for the conditions covered in these tests.

The correlation coefficient is an index which expresses the dependency of one variable on another when the relationship between them is assumed linear. The correlation coefficient can have a range of values from $+1.0$ through 0.0 to -1.0. A value of r equal to 1.0 indicates that all the observed values lie on the least squares line, *i.e.*, the points lie in a straight line when one variable is plotted against a second variable. An r value equal to zero indicates complete independence of the 2 variables. The least-squares line in this case is parallel to the abscissa. A negative correlation coefficient indicates that the dependency is opposite to that assumed in calculating the correlation coefficient.

For college age females exposed for 60 min. to floor-panel-heated environments, the comfort votes (using the scale given under the heading "Experimental Routine" of this paper) at the end of the exposure period are given in Table 3. The correlation coefficients for the correlation of mean comfort vote with the various parameters, as just listed, are given in Table 4.

†*Editor's Note*: The whole subject of comfort-in-air is being studied, and a project is under way at the Society's Laboratory to re-evaluate the comfort charts.

During January of 1957, a series of tests was undertaken using a constant air temperature of 75 F with floor temperatures of 80, 85, 90, 95 and 100 F. The results obtained from these tests are tabulated in Table 5. In addition to the final comfort vote, the final foot temperatures are recorded. These values are the means for the 27 subjects. Statistical analysis* of the data from this experiment showed that over a range of floor temperatures of 80 to 95 F for subjects seated at near basal conditions, the differences in comfort vote were not statistically significant and on this basis the differences were not due to the variation in floor temperature. The increase in comfort vote with a floor temperature increase from 95 to 100 F was statistically significant indicating that the subject could definitely detect an increase in floor temperature to 100 F.

The analysis of variance method[5] used in analyzing these data also showed that a quadratic equation would best fit the data indicating a sudden increase in vote with floor temperatures above 95 F.

TABLE 4—CORRELATION COEFFICIENT FOR COMFORT VOTE WITH VARIOUS PHYSICAL PARAMETERS FOR COLLEGE AGE FEMALES, SEATED AT REST, 60-MIN EXPOSURE 1953-54 TESTS

WOMEN 1953 TESTS COMFORT VOTES WITH	COEFFICIENT
Air Temp	$r = 0.93$
Floor Temp	$r = 0.22$
Effective Temp	$r = 0.93$
Operative Temp	$r = 0.95$
Air Temp + MRT	$r = 0.94$

Analysis of the foot temperatures was not as conclusive as the analysis of the vote data. However on a statistical basis, the significant differences in foot temperatures occur between floor temperatures of 85 to 90 F and between 95 to 100 F for a constant air temperature of 75 F. There was no definite correlation between foot temperature and comfort.

Statistical analysis of the 1950-51 tests using male students and the 1953-54 tests using female students led to conclusions similar to those previously given. For both male and female subjects varying floor temperatures did not significantly affect comfort votes when the air temperature was 75 F. The range of floor temperatures for these tests was 75 to 95 F. With the air temperature 5 to 10 F higher or lower than 75 F, the comfort votes were affected only slightly.

DISCUSSION

The heat transfer from and to the human body when exposed to various environments is affected by 4 major environmental factors, namely, air temperature, mean radiant temperature, relative humidity and air motion. During this investigation the latter 3 factors were either held constant or were shown to have negligible effect on the comfort votes obtained.

The air motion was held below a minimum value at the occupied level so that the effect of this factor on the results was the same for all test conditions. The

*Statistical work performed by the Kansas State College Statistical Laboratory.

Table 5—Comfort Vote and Foot Temperatures at the End of a 60-min Exposure for College Age Males, Seated at Rest at Air Temp 75 F

Floor Temp	Final Vote	Final Foot Temp
80	3.750	87.9
85	3.746	87.8
90	3.696	91.89[a]
95	3.836	91.9
100	4.027[a]	93.3[a]

[a] Increase statistically significant.

effect of relative humidity on the comfort votes has been discussed. As a check on the assumption concerning relative humidity, the correlation coefficient for the correlation of comfort vote with relative humidity was calculated. A value of 0.0688 was obtained indicating negligible correlation.

The mean radiant temperature affects the heat transfer by radiation to and from the body. In these investigations the MRT variation was due mainly to the floor surface temperature variations, one of the parameters of these tests. The effect of the floor temperature is therefore an indication of the effect of a variation in MRT for a constant air temperature except that the warm floor will also supply heat to the feet by conduction and will thereby raise the foot temperature. A variation in floor surface temperature of 10 F results in an increase in MRT, measured at the center of the room, of approximately 4 F assuming all other temperatures in the psychrometric room remain constant.

The data which are tabulated in Table 2 are shown in Fig. 1. The *least-squares* line[8] has been computed and is shown on the graph. Although the calculated line exhibits a slight variation of comfort vote with floor temperature, the statistical analysis shows that this variation is not significant, *i.e.*, the variation shown is probably due to variations within the sample and not due to true variations. This conclusion is further supported by the calculation of the correlation coefficients.

The data for the female students have not been shown since each air temperature has but 3 floor temperatures associated with it. However, the statistical design of the experiment was such that the data could be treated and the results led to similar conclusions.

Correlation coefficients of 0.90 or greater shown in Tables 1 and 4 indicate that the comfort vote can be considered a function of air temperature or of any of those

Table 6—Foot-finger Temperature Difference for Female Subjects Seated at Rest[a]

Floor Temp Fahr	Air Temp Fahrenheit		
	65	75	85
75	−0.33	−3.04	−5.36
85	+6.23	−0.70	−1.99
95	+7.12	+0.16	−1.14

[a] Test period 60 min.

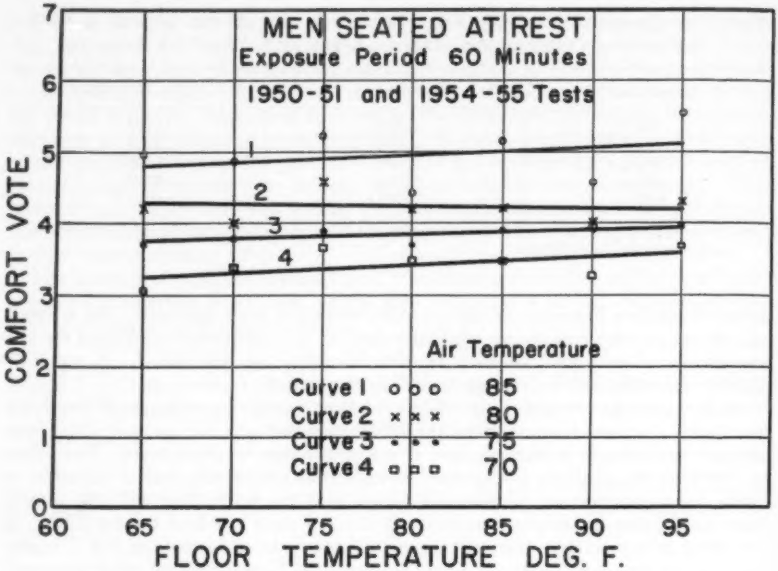


FIG. 1—EFFECT OF FLOOR SURFACE TEMPERATURE ON COMFORT

parameters in which air temperature is a major factor. These data also show that the comfort vote is affected little if any by the floor surface temperature over a range of floor temperatures of 75 to 95 F. These conclusions are limited to the conditions of these tests.

The effect of floor temperature on foot temperatures was found to be variable among the various subjects even when the subjects were wearing similar footwear. The footwear for the male students usually consisted of cotton socks and leather oxfords. For the female students, the footwear consisted of cotton anklets with loafers or saddle shoes, no high heeled shoes. For most of the experiments, the subjects were not subjected to a formal pre-test adaptation period. However, for the majority of the tests, and in particular those tests conducted in 1957, the subjects reported to the test room from classes in the engineering building so that the majority of the subjects had been adapted to the environment maintained for the building.

The data given in Table 5 are considered representative and indicate foot temperatures of the order of 88 to 91 F for subjects who are comfortable, seated at rest in an environment for 60 min. The temperatures given are the means of the temperatures recorded at the end of the test period. These temperatures are 1 to 4 deg higher than the temperature of 87 F suggested by Marechal as the maximum for comfort.

The foot-finger temperature difference was investigated during the tests using female subjects. The data were analyzed to determine if a relationship might exist between comfort vote and foot-finger temperature difference. The variation of results between subjects was inconstant and no conclusions could be formulated.

The means of the data did exhibit a pattern which is shown in Table 6 indicating a positive foot-finger temperature difference for a warm floor and cool air, and negative values for a cool floor and warm air. In warm air the skin temperature is increased to increase the heat loss and in cool air the opposite effect occurs. With a warm floor the foot temperature would tend to increase and with cool floors the foot temperature would tend to decrease.

It should be emphasized that the data and conclusions are limited to the test conditions maintained during the first phase of this program, namely, subjects seated at rest with an exposure period of 60 min. The second phase of this program will be undertaken to determine the effect of floor surface temperature on comfort for subjects seated at rest when the period of exposure is 3 hr. Also a study will be undertaken to determine the effects of floor temperature on standing subjects performing light work such as the housewife at the kitchen sink or the lathe operator in a factory.

CONCLUSION

Using 108 male and 21 female college students, the effect of floor surface temperature on the comfort votes was found to be negligible in a range of floor temperatures of 65 to 95 F when the subjects were seated at rest in an environment with a 75 F air temperature and with a period of exposure of 60 min. During the 1957 tests, using male students, it was found that the floor temperature of 100 F influenced the comfort vote although the actual value of the vote indicated comfort. From this information it is concluded that 95 F is the maximum for comfort under the conditions of these tests.

Statistical analysis of the data shows that the comfort vote could be correlated with air temperature, operative temperature, effective temperature and air temperature plus mean radiant temperature. This is mainly due to the use of air temperature in each of these parameters and, for the conditions used in these tests, the air temperature is the predominate factor influencing the subjects' reaction to the environments. Analysis of the 1957 data show significant differences among subjects. These differences could be accounted for in the analysis used and the conclusions given are justified.

Foot temperatures recorded at the end of the 60-min exposure period indicate that 88 to 91 is the maximum foot temperature for comfort under the conditions of these tests.

The results of these investigations cannot be used to predict the effect of floor temperatures on comfort when the period of exposure or occupancy is greater than 60 min or when the occupants are not seated with the body at near basal conditions. It is hoped that the second phase of this program will provide data for exposure periods of 3 hr with the occupant seated or standing doing light work.

ACKNOWLEDGMENT

The authors wish to acknowledge the assistance and encouragement in carrying out this project received from Prof. Linn Helander, Mechanical Engineering Department, Kansas State College.

REFERENCES

1. HEATING, VENTILATING AIR CONDITIONING GUIDE 1956, Chapter 24—Panel Heating, p. 581 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.).

2. Effects of Panel Location on Skin and Clothing Surface Temperature, by L. P. Herrington and R. J. Lorenzi (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 223).

3. Heated Floors and Comfort, by F. A. Chrenko (*Journal of the Institution of Heating and Ventilating Engineers*, Vol. 23, January 1956).

4. Permissible Temperatures for Ceiling and Floor Heating, by J. C. Marechal (*Journees Internationales de Chauffage, Ventilation et Conditionnement d'Air*, May 1955; in French).

5. Environment, Comfort, Health and People, by M. K. Fahnestock and J. E. Werden (*Refrigerating Engineering*, February 1956).

6. ASHVE RESEARCH REPORT No. 1380—Physiologic Examination of the Effective Temperature Index, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., and M. K. Fahnestock (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 51).

7. *Statistical Methods*, by G. W. Snedecor (4th Ed., The Collegiate Press, Ames, Ia., 1945).

8. *Advanced Mathematics for Engineers*, by H. W. Reddick and F. H. Miller (2nd Ed., John Wiley & Sons, Inc., New York, N.Y.)

DISCUSSION

W. P. CHAPMAN, Milwaukee, Wis., (WRITTEN): The authors have attacked a problem that has bothered designers for as long as floor-panel-heating systems have been used. For that reason alone, this paper is worthy of comment, and the authors should be given an acknowledgment of appreciation.

After studying this paper my curiosity was aroused on two points. One, does exposure time affect the comfort vote, and two, is comfort an adequate index.

The first point, concerning the exposure time is raised because an exposure time of one hour is not typical for occupants of schools, offices, factories or homes. It would be typical, however, for customers in stores, or other such buildings. If, then, we consider an exposure time of 4 hours as more typical would the votes of the test subjects have a different statistical pattern? For example, if a person has been outside for an hour in a temperature of 10 F and enters a building that is maintained at 60 F, his immediate reaction is one of comfortable warmth. After a few minutes, however, certain physiological changes have occurred and he senses that 60 F is not as comfortable as his first reaction indicated. It seems likely, therefore, that there is some minimum time before which we should assume the test subjects' vote is representative of a permanent sensation. I would be interested in hearing the authors comments on this point.

The second point that comes to mind is the index the designer or investigator should use in fixing the heated floor surface temperature. All will agree that comfort is one of the conditions we should consider, but also we realize how difficult it is to define comfort. In other words, comfort is a necessary condition, but is it sufficient?

Take the case of office workers. They must be comfortable, but also they must be alert. Herrington and Lorenzi—as the authors point out—considered vascular dilation in the legs and feet as an index of alertness. They found that floor surface temperatures above 75 F, although comfortable, induced vascular dilation. One would infer, then, that occupant's would be comfortable but inefficient with floor surface temperatures above 75 F. If that is so, are we exaggerating the role of comfort? Should we, instead, consider comfort *plus* some physiological index when establishing the optimum floor surface temperature? In addition, it would seem proper to determine what physiological index to use. Vascular tone has been popular, but then metabolic rate or respiratory rate might be simple to measure and reliable to use.

ALDO GINI, Milano, Italy (WRITTEN*): *Letter to W. J. Collins, Jr.*—As an old member of ASHAE, I have read the article by Messrs. Nevins and Flinner. I had previously read an article by F. A. Chrenko in the January 1956 issue of the *Journal* of the *Institution of Heating and Ventilating Engineers*, and following receipt of an invitation to

send a discussion, I sent some remarks which were published in the March 1957 issue of the *IHVE Journal.* Copy of these remarks is enclosed, also the reply of Mr. Chrenko, together with copy of a letter which I afterwards sent to B. A. Hodges, secretary of the English *Institution.*

As you see, the results presented by Nevins and Flinner are in full agreement with the interpretation I had given in the year 1938, and I think it would be interesting if in the next experiments they are announcing they will consider the separate influence of the other parameters (as the clothing, the air velocity, activity) that may influence the general heat exchanges of the human body and the metabolism.

I regret that owing to my advanced age I am not able to attend the 64th Annual Meeting of the Society, but I shall be grateful if you will communicate the content of my remarks during the discussion at the meeting.

P. S.—If you compare the figures of my table (see Table A) with the results of Nevins for the same room temperature, he indicates a higher floor temperature; this may

TABLE A—RELATION BETWEEN ROOM AND FLOOR TEMPERATURES AS GIVEN BY GINI

ROOM TEMPERATURE, FAHRENHEIT	FLOOR TEMPERATURE, FAHRENHEIT
58	94
62	90
66	86
70	83
74	81

perhaps be explained by the fact that in Italy we wear heavier winter clothing than in the United States.

Mr. Gini's Discussion of the Chrenko Paper—Mr. Gini felt that in view of the evidence existing from the work of Cadiergues and the experiments of Missenard (who found that in a church with an air temperature of 39 F, a floor temperature of 104 F was comfortable) that the importance of air temperature on maximum floor temperature cannot be denied. In his paper, published in 1938, *Il riscaldamento dal pavimento del puto di vista ignienico,* Mr. Gini had pointed out that in judging the allowable floor temperature in the physiological zone, three temperatures should be considered: (a) the cold zone, or vasomotor regulation; (b) the neutral point; and (c) the zone of evaporative regulation. The comfort zone lies on each side of the neutral point. Based on this thinking, Mr. Gini had given Table A to show the relation between room temperature and maximum floor temperature which he considered suitable for Italian conditions in winter, in normally built dwellings in Italy, and with normal Italian clothing.

Mr. Chrenko's Reply to Mr. Gini—Mr. Chrenko agreed that air temperature should be taken into consideration and also called attention to the curves of Marechal appearing in the October 1955 issue of *Industries Termiques,* where floor temperature is related to the percentage incidence of discomfort among men and women. Mr. Chrenko

---

**Editor's Note*: The written discussion of Aldo Gini consisted of a letter written to W. J. Collins, Jr., then chairman of the Publication Committee, ASHAE, and dated November 26, 1957. Portions of this letter are what appears here in the discussion under the heading of *Letter to W. J. Collins, Jr.* With his letter to Mr. Collins, Mr. Gini enclosed a copy of the written discussion which he had presented to a paper prepared by F. A. Chrenko entitled *Heated Floors and Comfort* in the January 1956 issue of the *IHVE Journal,* London. A shortening of that material is what appears here as *Mr. Gini's Discussion of the Chrenko Paper.* Included also in the letter which Mr. Gini sent to Mr. Collins was a copy of Mr. Chrenko's reply to Mr. Gini, and what appears here as *Mr. Chrenko's Reply to Mr. Gini,* is a shortening of that reply. Similarly what appears here as *Mr. Gini's letter to Mr. Hodges* is a condensation of a letter which Mr. Gini wrote to Mr. Hodges and copy of which letter was included with Mr. Gini's letter addressed to Mr. Collins.

stated that his results take various factors into account including activities, while Mr. Gini's table seems to give the same floor temperature for all activities.

Mr. Gini's Letter to Mr. Hodges, November 1957—In this letter, Mr. Gini reiterated his belief that maximum floor temperature depends on various parameters of which air temperature is one. He thought that experiments ought to be carried out and compared in a logical, theoretical frame and stated that he thought the most convenient frame is given by the position of the subject in the zone of the heat exchanges. He also mentioned that he felt that the experimental results of Nevins and Flinner confirmed Mr. Gini's own conclusions as stated in 1938.

L. P. HERRINGTON*, New Haven, Conn., (WRITTEN): This careful study of the 60-minute effect on thermal comfort votes of floor temperatures from 65 to 95 F at air temperatures from 65 to 85 F, demonstrates that in young individuals at rest the effect of floor temperature on voted comfort is small.

The authors are guarding against casual interpretation of this clear result as a general clearance for floor temperature of the order of 85 to 95 F by extension of their valuable studies to a 3-hour exposure, and they plan tests in which the subjects will stand, and engage in activity typical of domestic or other work.

The chief issue of thermal hygiene raised by this study and future studies of the same general plan is whether or not it is possible to evaluate by thermal comfort votes the subjective effect of longer term relaxant circulatory changes resulting from locally higher foot temperatures (Table 5) if these are maintained over a complete season. As a secondary issue, it will be important to know whether results on young subjects with high vascular tone can be safety applied to a population with a mean age of about 40 years.

What is established in this field can be stated briefly. With their large surface-to-volume ratio, the extremities are excellent heat exchangers. Substantial variations in extremity temperature represent a highly important means of continuous adjustment to various temperatures, and to changes in activities, at a standard temperature. This important adaptive ability is in part dependent upon floor contact of the feet with surfaces at temperatures below the foot temperatures of 80 to 90 F which accompany this adaptation.

It is clear from Table 5 of this paper that at an air temperature of 75 F, increasing floor temperatures have a much larger effect on foot temperature than on 60-minute comfort votes. At 90 and 100 F floor temperatures, the foot temperatures approach the 92 to 95 F foot temperature associated with general vasodilation and summer conditions of skin temperature acclimatization. In the absence of definitive information on the relation of vascular tone, alertness, and voted thermal comfort, the present reviewer prefers heating arrangements which at a general temperature of 70 F leave the extremities free to vary between 80 F and 90F, thus preserving a body heat balancing device, which is lost if local floor temperatures force the extremities to the higher temperatures typical of vasodilation. (See Reference 2 of the paper).

It would seem, therefore, that the problem which Professors Nevins and Flinner are examining in such a thorough manner has two central questions re thermal hygiene:

(a) Is the strictly thermal comfort of the individual disturbed by local floor temperatures of the order of 85 F to 95 F, if the overall heat loss is balanced? It seems possible that the answer to this question may be negative. Particularly so if the general comfort level maintained is a slightly cool comfort in which the rise in foot temperature is held to a minimum, and the exposure time is of the order of an hour.

(b) In the second instance, we may ask: What are the long term seasonal effects on alertness and postural tone if foot temperatures not specificially felt as uncomfortable, are such as to induce a vasodilation normally associated with the relaxant effects of warmer temperatures? Further, since compensation for heat induced changes in vascu-

* John B. Pierce Foundation.

lar tone is better in young subjects, should an attempt be made to answer the problem for the mean population age of circa 40 years; and the older half of the population, in whom vascular relaxant effects may have adverse effects on circulatory adjustments to postural change and general body tone?

This reviewer hesitates to suggest the addition of further types of observation to this very full program of Professors Nevins and Flinner. However, it is probable that the association of a physiologist with this program, to measure circulatory adjustment to postural change following the 3-hour exposures, would do much to give their valuable results a wide application to the problems of thermal hygiene.

R. P. SCHOENIJAHN, Wilmington, Del.: I would like to ask the authors if any study has been made with regard to comfort temperatures for *barefoot* conditions. Radiant heat facilities as in Hospital for Mental Cases have been installed where the patients frequently walk around barefoot—this also applies to certain Gymnasium Locker Room areas where there are concrete floors on grade with no covering.

There has been some discussion as to what floor temperature should be maintained in view of supplementary out-door air supply which is necessary in locker rooms. Have any of the *panel* procurred any data or do they have suggestions on these phases of design?

D. R. BAHNFLETH, Urbana, Illinois: One minor question occurred to me when reviewing the paper. It was noted that the room was cooled with conditioned air supplied through ventilation panels located in the ceiling. It would seem to me that if the ventilation panel area were large with respect to the area of the ceiling, and if the conditioned air supply were constant throughout the investigation, that the average ceiling-surface temperature would decrease with increases in floor-surface temperature. For example, if we consider a room-air temperature of 70 F and a floor temperature of 95 F, the space sensible cooling load would be approximately 4,000 Btuh, whereas with a 70 F room temperature and 70 F floor temperature, the space cooling would be negligible. Thus, if the air flow were constant the average ceiling-surface temperature would be lower in the first example cited.

I would like to ask whether data are available that would indicate how much the ceiling-surface temperature would decrease with increased floor-surface temperature and to what extent any decrease in ceiling-surface temperature would affect the comfort sensations indicated by the subjects.

M. K. FAHNESTOCK, Urbana, Illinois: I am quite interested in this paper because it pertains to human comfort and especially because Curve 3 in Fig. 1 indicates that with an air temperature of 75 F the subjects were comfortable with a range of floor temperature of from 65 to 95 F. This is a little surprising and it may be that the exposure of one hour was not long enough for the subjects to become thoroughly adjusted to the given environments. Perhaps at the end of two or three-hour exposures to the various conditions the comfort votes would have been different than they were at the end of one hour. Dr. Herrington pointed this out in his written discussion.

As the authors know, observations of rectal temperature, regional skin temperatures and mean skin temperature will show when an individual is in thermal equilibrium with his environment. Although taking regional skin temperatures greatly complicates the test procedure doing so gives objective data which are very important in human comfort studies and makes possible the calculation of mean skin temperatures which have been correlated with subjective comfort votes by several investigators. Thus, this procedure gives an objective measure of comfort.

The other thing which I want to bring out is that in order to remain comfortable in an office, home, or similar room the human body makes certain physiological adjustments. The heat production varies with activity, such as sitting, standing, or walking and with food intake. Thus, in order to remain comfortable in a given environment an individual's heat loss to his surroundings must vary within limits with his heat

production. The human body, largely by changing the temperature of its extremities, the hands, arms, legs and feet can within limits regulate its heat loss and thus maintain a balance between heat production and heat loss.

For example, in maintaining this balance in an environment ranging from 73 to 77 F with the surface temperatures, including that of the floor within a few degrees of the air temperature, the temperature of the feet of a sedentary individual may vary some 4 degrees, say, from about 83 to 87 F, while the mean skin temperature of the entire body may vary only about 1 degree, from 90.5 to 91.5 F. Thus, the extremities the feet, legs, hands and arms are the portions of the body which make the major adjustments to keep the heat production and heat loss balanced. The question is, does a panel-heated-floor seriously interfere with this adjustment capacity of the lower extremities. Based on comfort votes the results presented in Fig. 1 indicate that it does not, at least, for exposure periods of one hour. It would be very interesting to see what complete skin temperature data would show for the one hour and for longer exposure periods.

F. J. LINSENMEYER, Detroit, Michigan: It seems that the test group reporting comfort conditions differ greatly from past experience but I would imagine there will be some explanation forthcoming to correlate with previous tests in this field. I wonder whether the authors are prepared to offer some explanation for the high surface temperatures. We have a very wide departure from the practice of the last twenty years.

AUTHORS' CLOSURE (Mr. Nevins): The authors wish to express their appreciation for the many comments and suggestions, particularly those by Dr. Herrington, Professor Fahnestock and Professor Gini. The authors have pointed out in the paper that the results of the first phase of this program cannot be used to predict the effect of floor temperatures on comfort when the period of exposure is greater than 60 min or when the occupants are not seated with the body at near basal conditions. Nor can the results be interpreted as design criteria.

In regard to Mr. Chapman's questions concerning exposure time, the authors anticipate that the subjects will react differently to periods of exposure of three or four hours. The second phase of these tests will be conducted using 3-hour periods in an attempt to determine these differences. The data used in the paper are the comfort votes obtained at the end of the 60-min exposure. The second point raised by Mr. Chapman concerns the proper index to be used to define comfort. Dr. Chrenko prefers the use of local comfort votes while the authors have used the comfort sensations of the whole body. The whole problem of assessment of subjective reactions and the feeling of comfort is an important one and considerable thought is being given to it.

The authors have not made any investigations with barefoot subjects. The reaction of the subjects would be quite different and the flooring material would influence greatly the results.

In reply to Mr. Bahnfleth, the cooling air supply was not constant so that the surface temperature of the air panels in the ceiling varied during the tests. These panels comprise 40 percent of the ceiling and would produce a little radiant cooling. The data show temperature swings of approximately 2 F deg. An attempt will be made to evaluate this effect during the second phase test program.

The authors are indebted to Professor Gini for his interesting observations and comments. Professor Gini's general agreement with the findings presented in the paper is gratifying.

The comments by Dr. Herrington and Professor Fahnestock, both of whom have contributed much to physiological and comfort research, will be incorporated in future research carried out by the authors. It is hoped that mean skin data for subjects exposed for three hours can be obtained during the second phase tests and the results presented at a later date. An attempt will also be made to obtain data regarding the circulatory adjustments of the subjects.

1631

No. 1631

LIGHTING AND COOLED AIR EFFECTS ON PANEL COOLING

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

By L. F. SCHUTRUM* AND T. C. MIN**, CLEVELAND, OHIO

MORE THAN 20 percent of the energy supplied to fluorescent lights can be removed from a room by a cooled ceiling-panel at 70 F. This and other interesting observations were made in the Society's Environment Laboratory[1] where experimental work on panel cooling was in progress from the middle of 1954 until early in 1956, under the guidance of the TAC on Panel Heating and Cooling.

An earlier paper[2], published in January, 1955, presented data on the heat removed by a cooled ceiling-panel when the only heat sources were the warm walls and floor of the room, and warm infiltration air. This paper presents the results of later studies made to determine the effects of chilled air introduction and direct and indirect fluorescent lighting on the heat pickup of a cooled panel. Ninety-five tests were made to obtain the information reported here. In one test, 8 staff members occupied the room; in all others, the room was unoccupied, and therefore, no latent heat load existed. In all tests, the room was unfurnished.

TEST ROOM

All 6 surfaces of the room[2] were composed of aluminum panels through which liquid at any desired temperature could be circulated. Surface temperatures were measured by copper-constantan thermocouples. Heat flows into or out of the surfaces of the room were measured by plate-type heat-flow meters[3] fastened to the room side of the aluminum panels. The ceiling was painted white, the walls green, and the floor gray. The room was 25 ft × 12 ft, with a ceiling which could be adjusted in height up to 12 ft. Conditioned air entered the room through

*Research Supervisor, ASHAE Research Laboratory, Member of ASHAE.
**Research Engineer, ASHAE Research Laboratory, Member of ASHAE.
[1]Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR CONDITIONING ENGINEERS, Pittsburgh, January, 1958.

two 8-in. ceiling diffusers and left through a slot between one wall and the floor. The quantity of air was metered by an orifice, and provision was made for introducing moisture into the conditioned air to maintain a relative humidity of 50 ±10 percent.

All test data were taken under steady-state conditions. In most tests, the floor and the 4 walls were maintained at the same selected uniform temperature between 70 and 95 F, thus creating a so-called *uniform environment*. This uniform environment temperature is also referred to as the AUST (area-weighted average un-

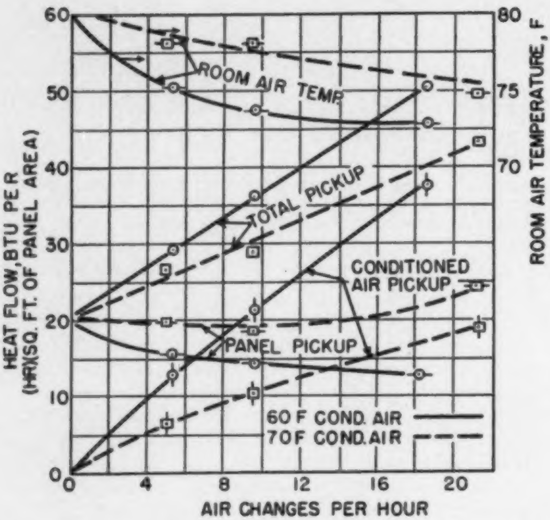


Fig. 1—Effect of Conditioned-Air Supply Rate and Temperature on Heat Removal by Panel and Air (Ceiling 70 F, AUST 85 F, 8-ft Ceiling Height)

cooled surface temperature). The entire ceiling was also held at some selected uniform temperature from 60 F to 85 F.

Effects of Introducing Cooled Air

The exact quantitative effects of cool air introduction on the performance of a cooled ceiling-panel may vary, depending upon location, type and adjustment of the air distributing device used. In the tests reported herein, air was supplied through two 8-in. ceiling diffusers located at the approximate centers of the 2 halves of the room, and adjusted to distribute the air horizontally. The diffusers extended approximately 2 in. below the ceiling, and the air stream did not immediately attain close contact with the ceiling. The size and adjustment of the diffusers were such that the throw of the air stream was less than the diffuser-to-wall distance at all air flow rates less than 2.1 cfm per sq ft of floor area.

Conditioned air was supplied at temperatures from 60 F to 75 F, and in quantities up to 21 air changes per hour (about 2.8 cfm per sq ft of floor area for an 8-ft ceiling height). The relative humidity in the room was held between 40 and 60 percent and for this part of the study, the room was unoccupied and unlighted.

Fig. 1 shows the observed effects of cool air introduction to a room having a uniform environment of 85 F and a cooled ceiling-panel at 70 F. Note that the heat pickup by the cooled air increased as the air quantity increased and as its temperature decreased. The heat pickup by the cooled panel remained relatively constant when the temperatures of the cooled air and the panel were the same, regardless of the air volume. However, in tests where the entering air temperature was below the panel temperature, the panel heat pickup decreased as the air volume increased.

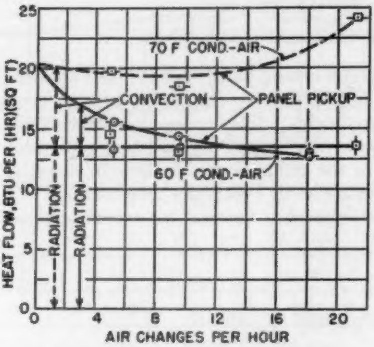


FIG. 2—RADIATION AND CONVECTION COMPONENTS OF TOTAL PANEL HEAT PICKUP (CEILING 70 F, AUST 85 F, 8-FT CEILING HEIGHT)

The combined heat removal of the air and panel, and the resulting room air temperatures are also shown in Fig. 1.

The panel heat pickup curves of Fig. 1 have been replotted in Fig. 2 and their convective and radiative components are shown. The calculated[4] radiant component is shown to be about 13.7 Btu per (hr) (sq ft) for the indicated surface temperatures. The convective component was determined by subtracting the radiant component from the observed total. As shown in the figures, when air at 70 F was supplied the cooled panel picked up an appreciable quantity of convected heat at all air-flow rates. However, when the temperature of the conditioned air was reduced to 60 F, the pickup of convected heat reduced rapidly as the air-flow rate increased, and became negative at about 13 air changes per hour.

The data reported in Figs. 1 and 2 were determined for an 8-ft ceiling height. Tests were repeated at 9½ and 11½ ceiling heights, and no significant changes in panel performance were found.

For a few tests, the diffusers were adjusted to direct the air vertically downward, and an appreciable increase in the heat pickup of the panel was noted. With conditioned air entering at 60 F, the panel heat pickup was approximately the same as shown in Fig. 2 for a 70 F conditioned-air temperature.

Heat transfer to a cooled surface by convection is usually calculated by the equation

$$q_c = h_c(t_a - t_p) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

where

h_c = coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree) based on $(t_a - t_p)$.
t_p = surface temperature, Fahrenheit.
t_a = room-air temperature, Fahrenheit.

In the case of a cooled ceiling-panel, with cooled air being distributed across it horizontally from a ceiling diffuser, Equation 1, with the foregoing definition of symbols, was not applicable. The room-air temperature differed considerably from the temperature of the cool air flowing across the ceiling, and, at times, the measured heat flow to the ceiling would be zero or negative in spite of a positive difference between the room-air and panel temperatures.

As explained in the Appendix, a second equation of the same form as Equation 1 was written,

$$q_c = h_e(t_e - t_p) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

but the symbols were defined as follows:

t_e = equivalent film temperature, Fahrenheit.
h_e = equivalent coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree) based on $(t_e - t_p)$.

With Equation 2, a zero value of $t_e - t_p$ indicated zero heat transfer. In other words, the equivalent film temperature when no heat flow occurs through the surface. By this concept, the equivalent film temperatures and equivalent coefficients of heat transfer were determined and correlated in terms of the AUST, the entering air temperature, and the rate of air supply. By substituting these values in Equation 2, it was possible to calculate the convection heat pickup by the ceiling panel for a variety of conditions. The radiant heat pickup for the same conditions was calculated by the Stefan-Boltzmann law.

In Fig. 3, the calculated heat pickup by the ceiling panel is plotted against the rate of flow of cooled air supply and the rate of heat removal by it, for various differences between room-air and ceiling-panel temperatures. The figure is based upon data obtained in tests in which the only heat gain to the space was from the warm floor and walls. However, the application of the figure to the solution of problems involving internal loads will be demonstrated later in the paper. The figure is for a supply air temperature of 60 F. It should be pointed out that the relationships shown in Fig. 3 are those observed in the Environment Laboratory. Application of the figure to other spaces has not been verified.

Note that for a given difference between room air and ceiling temperatures, heat removal by the panel is greater at higher rates of air flow. It was shown in a previous paper that when the only heat gain to the space was from the warm floor and walls, a cooled ceiling-panel could pick up approximately 2 Btu per (hr) (sq ft)

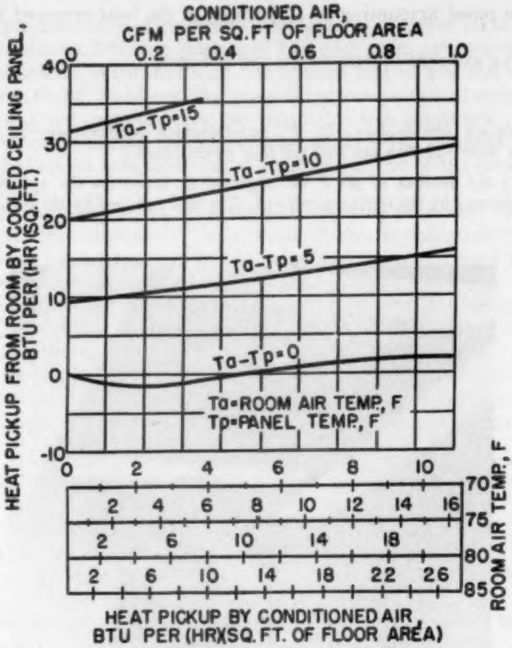


FIG. 3—HEAT PICKUP BY COOLED CEILING-PANEL AND CONDITIONED AIR AT 60 F (8-FT CEILING HEIGHT)

(deg difference between the room-air and panel temperature). If the heat gain is greater than this, the room temperature can be held constant only by the introduction of cooled air. For such a condition, the following equation, based upon the curves of Fig. 3 for 5 and 10 deg temperature differences, can be written to express the approximate heat pickup by the panel:

$$q_p = 2(t_a - t_p) + 0.3\,[1.08C(t_a - t_{ca})] \quad \ldots\ldots\ldots \quad (3)$$

where

- q_p = heat pickup by the ceiling panel, Btu per (hour) (square foot).
- t_a = room-air temperature at the 60 in. level, Fahrenheit (limited to 75 to 85 F).
- t_p = panel-surface temperature, Fahrenheit (limited to 70 to 75 F, but lower than t_a).
- t_{ca} = conditioned-air temperature, Fahrenheit (limited to 60 to 70 F).
- C = rate of conditioned air supply, cfm per square foot of floor area (range 0 to 1.0).

The first part of Equation 3 is the previously reported heat pickup of 2 Btu per (hr) (sq ft) (deg temp difference). The second part indicates an additional heat

pickup by the panel amounting to 30 percent of the heat removed by the cooled air.

The following example illustrates the use of Fig. 3.

Example 1:

Given: *Room-air temperature—75 F; panel-surface temperature—70 F; heat gain from walls and floor—12 Btu per (hr) (sq ft of panel area).*

To find: (1) *the amount of 60 F air required to maintain the specified conditions;* (2) *The heat removed by the ceiling panel;* (3) *The heat removed by the conditioned air.*

FIG. 4—INTERIOR OF ENVIRONMENT LABORATORY SHOWING ARRANGEMENT OF FIXTURES FOR DIRECT LIGHTING

Solution: The difference between the room-air and ceiling temperature is 5 deg. Follow along the line for 5 deg temperature difference until the sum of the heat pickup by the panel, as read at the left, plus the heat pickup by the air, as read on the 75 F line at the bottom, is 12 Btu per (hr) (sq ft). This condition is satisfied at an air supply rate of 0.12 cfm per sq ft of floor area, and the heat removal by the panel and air are 10 and 2 Btu per (hr) (sq ft of panel area) respectively.

EFFECT OF INTERNAL CONVECTIVE HEAT SOURCES

The effect of low-temperature convective heat sources on the performance of a cooled ceiling-panel was reported in the previous paper[2]. Another series of tests was made to determine the combined effect on panel performance of such heat sources and cooled air introduction.

As in the previous case, each heater consisted of a 500-w incandescent lamp shielded by concentric, 2-ft long sections of 8-in. and 10-in. galvanized duct. Each heater was supported vertically, with the bottom end of the sheet metal about 12-in. above the floor. Baffles at the top and bottom prevented radiation directly from the lamp to the room surfaces, but permitted free circulation of air through the device. When operating the temperature of the outer cylinder was less than 100 F, and calculations indicated that at least 95 percent of the total heat output was by convection.

As would be expected, when an internal convective heat load was added, the room-air temperature increased. As a result of this temperature rise, the combined increases in the heat pickup by the panel and the conditioned air were sufficient to remove this additional convective load.

If the room-air temperature is to be held constant when an internal convective load is added, the ceiling temperature must be decreased, or the conditioned-air quantity must be increased or its entering temperature decreased. If the ceiling and entering air temperatures are held constant, the ceiling heat pickup will remain essentially constant and practically all of the additional internal convective load must be removed by increasing the quantity of conditioned air.

Example 2: Assume in Example 1 that the room-air, panel-surface, and entering-air temperatures, and the heat gain from warm surfaces remain as stated. Also assume an additional internal convective load of 5 Btu per (hr) (sq ft of panel area). Determine the quantity of conditioned air required, and the heat picked up by the air.

Solution: In this case, the heat removal by the panel will be the same as in *Example* 1, or 10 Btu per (hr) (sq ft). The heat removal by the conditioned air will be 17—10 or 7 Btu per (hr) (sq ft of panel area). From Fig. 3, it is found that this heat removal from a 75 F room with 60 F entering air will require 0.43 cfm per sq ft of floor area.

EFFECT OF FLUORESCENT LIGHTING

To determine the effect of lighting on the heat pickup by a cooled ceiling-panel, 13 RLM Type 6[5] fluorescent fixtures were installed in the Environment Laboratory. The fixtures were first installed as shown in Fig. 4, to provide direct lighting. Later they were inverted to throw the light upward and provide indirect lighting. In both cases the fixtures were installed with the center lines of the tubes 18 in. below the ceiling.

Each fixture held three 40-w standard cool-white fluorescent tubes, and the ballast in each fixture had an average full load loss of 23 w. Thus the normal rating for the tubes and ballast in each fixture was 143 w, and the total for the 13 fixtures was 1859 w. This is equivalent to approximately 6 w per sq ft of floor area.

Power to the lights was supplied through a variable transformer which was adjusted to maintain approximately 118 volts at the fixtures throughout the tests. In spite of this control, the wattage to the lights did not remain constant. This was probably due to 2 factors, (1) the characteristic of fluorescent lamps to vary the current flow with variations in bulb-wall temperature, and (2) slight inaccuracies in voltage control. Wattages or their Btuh equivalents given later in the paper are those determined for specific test conditions, unless otherwise indicated. The average lamp-wall temperatures, with fixtures mounted for direct and indirect lighting, were approximately 125 F and 110 F, respectively.

Light output of a fluorescent lamp also varies considerably, depending upon tube temperature. The effect of ambient-air temperature and air movement on

the relative light output of bare fluorescent lamps is given in Fig. 1 of Reference 6. The average level of illumination in the test room 30 in. above the floor, was 185 ft-candles for direct lighting, and 82 ft-candles for indirect lighting. These readings were taken with a multicell color-and-cosine-corrected light meter. Fluorescent-tube and fixture surfaces were dusted weekly to assure uniform test conditions.

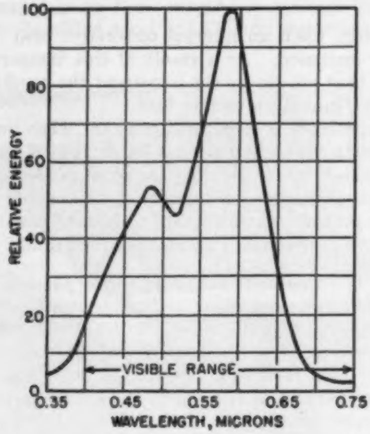


FIG. 5—SPECTRAL ENERGY DISTRIBUTION OF STANDARD COOL WHITE FLUORESCENT LAMP. MERCURY LINES 0.4047, 0.4358, 0.5461, AND 0.5780 MICRONS ARE EXCLUDED (FROM REFERENCE 9)

All of the electrical energy supplied to the lights was dissipated as heat by some combination of the following methods:

1. Radiation to the room surfaces;
2. Radiation to the carbon dioxide and water vapor in the room air;
3. Convection from the warm lamp and luminaire surfaces to the room air;
4. Conduction through the supporting members.

The low temperature radiation absorbed by the carbon dioxide and water vapor in the air was calculated by use of Reference 7, and was found to be unimportant. Gaseous absorption in the visible wavelengths is practically non-existent.[8] Heat conduction through the fixture supports was found to be insignificant. Therefore, for practical purposes, all of the heat transfer from the lights may be accounted for by radiation to the room surfaces or by convection to the room air.

The portion of the radiant energy from the lights which can be absorbed by the room surfaces depends upon the spectral distribution of the radiation at its source and the absorbing characteristics of the surfaces. The spectral curve for the

short-wave, high-temperature radiation from a 40-w standard cool-white fluorescent lamp is given in Fig. 5. This does not include the long-wave, low-temperature radiation given off by the warm surface of the tube. It will be noted from Fig. 5 that nearly all of the high-temperature radiation from the lamp is in the visible range.

The ceiling of the test room was painted with an off-white oil paint. The reflectance of this paint at various wavelengths, as determined by a recording spectrophotometer, is shown in Fig. 6. Absorptance of the surface for a given wavelength is determined by subtracting the reflectance for the wavelength from 1.00. Absorptance of the ceiling for the short wave radiation emitted by the lights, as

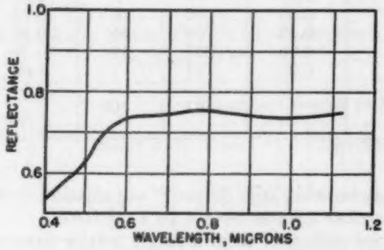


FIG. 6—SPECTRAL REFLECTANCE OF WHITE CEILING PAINT

obtained by an integration of the emitted energy (Fig. 5) and the ceiling absorptance (Fig. 6), was found to be 32 percent.

To learn more about the heat dissipation from the lamps, tests were made to measure the visible or short-wave radiation to each of the room surfaces. To accomplish this, the output of the heat-flow meters on one surface of the room was connected in series with a potentiometer which could supply an equal potential having opposite polarity. Any unbalance in the 2 voltages was then fed through an amplifier and read on an indicator. The lights were turned on and operated until temperature equilibrium was obtained, and the potentiometer was adjusted until the indicator showed no unbalance in the opposing voltages. When the lights were turned off, the high-temperature radiation stopped instantly, and the resulting drop in the heat-flow meter output was read. The reading was taken before the fixtures had cooled appreciably, but was delayed enough to allow the meters to respond.

The described measurements were made for each of the 6 room surfaces, for both direct and indirect lighting. Results are shown in Table 1. Note that the relative absorption of short-wave radiation by ceiling and floor differ widely for direct and indirect lighting. High-temperature radiation exchange between lights and room surfaces does not change appreciably with normal changes in surface temperature. The values shown in Table 1 were therefore judged to be applicable to all tests.

In addition to short-wave radiation directly from the light source, there was an appreciable amount of long-wave, or low-temperature radiation emitted from the surfaces of the tubes and from the warm surfaces of the fixtures. This low-tem-

TABLE 1—SHORT WAVE (LIGHT) ENERGY ABSORBED BY ROOM SURFACES RADIATION FROM THIRTY-NINE 40-WATT FLUORESCENT TUBES, TYPE RLM-6 FIXTURES

ROOM SURFACES	TYPE OF LIGHTING					
	DIRECT			INDIRECT		
	Btu per (hr.) (sq ft)	Btuh	percent	Btu per (hr.) (sq ft)	Btuh	percent
North Wall	0.39	46	8	0.25	29	5
South Wall	0.39	46	8	0.25	29	5
East Wall	0.39	96	16	0.25	61	10
West Wall	0.39	96	16	0.25	61	10
Floor	0.96	387	49	0.39	116	19
Ceiling	0.07	21	3	1.05	316	51
TOTAL		592	100		612	100

perature radiation may be arbitrarily defined[10] as radiation to which glass is opaque. Its wavelength is therefore greater than 4 to 5 microns.

The low-temperature radiation from a single 3-tube fixture was studied in the test room and a method of calculating this component of the heat exchange was developed, and is described in the Appendix.

With the 2 parts of the radiation component available, either by measurement or calculation, the convection component of heat transfer from the lights was determined for each test by subtracting the total radiation component from the total energy supplied to the lamps. The convective heat transfer obtained by this method was checked by making a convection heat balance on the room, *i.e.*, the convected heat added to the room from the surfaces and from the luminaries, must

TABLE 2—DISTRIBUTION OF ENERGY FROM FLUORESCENT LIGHTS TO ROOM SURFACES AND AIR

TYPE OF LIGHTING	ENERGY INPUT TO LIGHTS BTUH	ENERGY RELEASE FROM FLUORESCENT LIGHTS, PERCENT		
		SHORT WAVE (LIGHTS) RADIATION[a]	LOW TEMPERATURE RADIATION	CONVECTION
	CEILING 70 F, WALLS & FLOOR 80 F, 5 AIR CHANGES AT 60 F			
Direct	6150	9.6	31.9	58.5
Indirect	6467	9.5	21.5	69.0
	RANGE OF VALUES FOR VARIOUS TEST CONDITIONS			
Direct		9.0–9.8	24.3[b]–39.0[c]	51.6[c]–66.4[b]
Indirect		8.9–9.7	14.8[b]–24.1[d]	66.4[d]–75.6[b]

[a]Assumed constant for all tests; [b]10 air changes per hour; [c] 75 F conditioned air; [d]No conditioned air.

equal the convection loss to the room surfaces and to the conditioned air. The results obtained by the 2 methods differed by less than 5 percent for all but one of the 20 tests.

The distribution of energy transferred from the lights to the room is given in the upper part of Table 2 for a specific set of room conditions. The percentages of the high-temperature radiation to the room surfaces were calculated from the totals in Table 1, and are approximately the same for direct and indirect lighting. As might be anticipated, the convection was somewhat less with the reflector turned downward for direct lighting, than in the inverted position of the indirect luminaire.

TABLE 3—PICKUP OF RADIANT ENERGY FROM FLUORESCENT LIGHTING BY CEILING PANEL AT 70 F

Type of Lighting	Energy Input to Lights Btuh	Radiation Absorbed by Panel			
		Short Wave (Light)		Long Wave (Low Temperature)	
		Btuh[a]	Percent of Total Input to Lights	Btuh	Percent of Total Input to Lights
		Walls & Floor 80 F, 5 Air Changes (0.79 cfm per sq ft) at 60 F			
Direct	6150	21	0.3	938	15.3
Indirect	6457	316	4.9	866	13.4
		Range of Values for Various Test Conditions			
Direct		21	0.3	507–1209	8.31–9.7
Indirect		316	4.9	526–1054	8.11–6.3

[a]Values from Table 1.

The lower part of Table 2 gives the range of values obtained under various test conditions and the numerical values would undoubtedly vary for different types of fixtures.

PICKUP OF HEAT FROM LUMINAIRES BY COOLED CEILING-PANEL

Of primary interest to the engineer is the amount of heat from the lights that can be absorbed and removed by a cooled ceiling-panel. The upper part of Table 3 presents data on the radiation from the luminaires to the panel for 2 specific tests. These are the same 2 tests for which data are given in Table 2. The second part of Table 3 shows the range of values determined in other tests for other air change rates and inlet air temperatures.

Figs. 7, 8, and 9 show the effects of 3 different variables on the heat removed from the test room by the cooled panel and by the conditioned air. The sources of the heat were the warm wall and floor surfaces and the fluorescent lights. Results are given for both direct and indirect lighting systems.

Fig. 7 shows that as the volume of conditioned air increases, the heat pickup by the panel decreases, and the heat removal by the conditioned-air increases. It is evident from Fig. 8, that the panel pickup increases and the heat removal by the air decreases as the supply-air temperature increases.

Fig. 9 shows that as the panel-surface temperature increases, the heat removal by the conditioned air increases, and the heat pickup by the panel shows a rapid

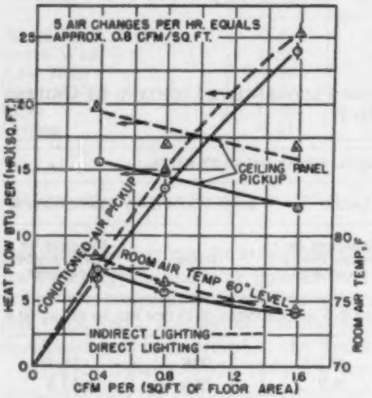


FIG. 7—EFFECT OF CONDITIONED-AIR SUPPLY RATE ON HEAT PICKUP BY COOLED CEILING PANEL AND CONDITIONED AIR; (FLUORESCENT LIGHTING 6 W PER SQ FT, CEILING 70 F, AUST 80 F, CONDITIONED AIR 60 F, 10-FT CEILING HEIGHT)

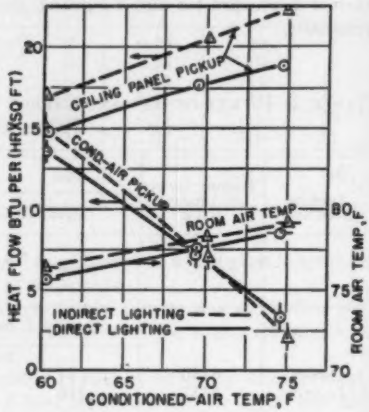


FIG. 8—EFFECT OF INLET AIR TEMPERATURE ON HEAT PICKUP BY COOLED CEILING PANEL AND CONDITIONED AIR; (FLUORESCENT LIGHTING 6 W PER SQ FT, CEILING 70 F, AUST 80 F, 0.8 CFM PER SQ FT, 5 AIR CHANGES PER HR, 10-FT CEILING HEIGHT)

decrease. At surface temperatures above about 80 F, the panel added heat to the space.

It is interesting to note that in each of Figs. 7, 8, and 9, the heat removal by both the panel and the conditioned air is greater for indirect than for direct lighting, but that in spite of this, the room-air temperature is higher with indirect lighting. This unusual relationship is the result of two factors: (1) the energy input was slightly higher for indirect than for direct lighting, and (2) the major part of the radiation from the direct lighting fixtures fell on the warm floor and wall surfaces, and while their temperature was increased somewhat, the total heat release from these surfaces to the cooled panel and the room air was less than the sum of the radiation received from the lights and the release which would have taken place from the surfaces at their normal temperatures. Baker[11] and others have indicated that absorption of radiant energy by the enclosure surfaces causes a higher inside-surface temperature and, therefore, reduces the heat gain by conduction.

The effect of the rate of supply of conditioned air on the various components of heat pickup of a cooled panel is shown in Fig. 10. As might be expected, radiant components are relatively unaffected by the conditioned air, but the convection component decreases as the air delivery rate increases.

In a room such as the Environment Laboratory, where heat is supplied from warm building surfaces and a lighting system, and cooling is accomplished by a

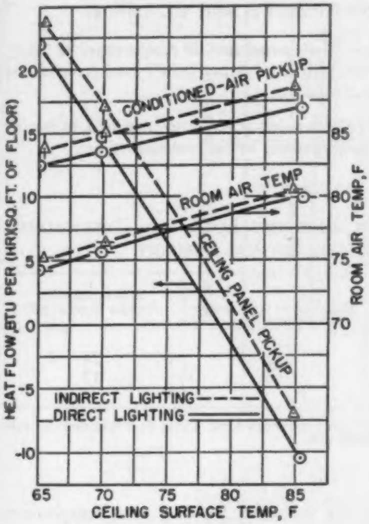


FIG. 9—EFFECT OF CEILING SURFACE TEMPERATURE ON HEAT PICKUP BY COOLED CEILING PANEL AND CONDITIONED AIR; (FLUORESCENT LIGHTING 6W PER SQ FT, AUST 80 F, CONDITIONED AIR 60 F, 10-FT CEILING HEIGHT)

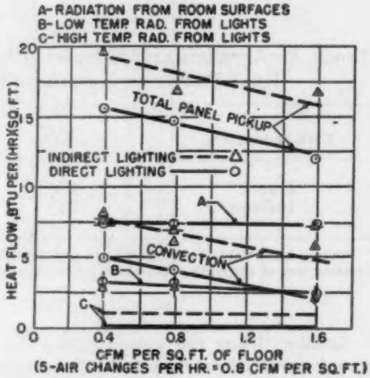


FIG. 10—EFFECT OF CONDITIONED-AIR SUPPLY RATE ON COMPONENTS OF CEILING PANEL HEAT PICKUP; (CEILING 70 F, AUST 80 F, CONDITIONED AIR 60 F, FLUORESCENT LIGHTING 6W PER SQ FT, 10-FT CEILING HEIGHT)

cooled ceiling and conditioned air, the heat exchange which takes place is the result of many interrelated factors. Although precise determinations would be extremely difficult, Fig. 3 may be used to predict the approximate heat removal by the panel and the conditioned air, over the range of conditions studied. The method of solution may be summarized as follows:

1. The normal heat gain through the room surfaces plus the radiation from the lights which falls on the floor and walls, may be treated as described in *Examples* 1 and 2 for the heat gain from the wall surfaces.

2. The convection heat gain from the lights may be considered an internal convective load and treated as described in *Example* 2.

3. The total radiation from the lights to the panel may be added to the panel pickup as determined from Fig. 3.

An approximate distribution of lighting load suggested for use in making a solution by means of Fig. 3, is given in Table 4. The values in this table are based on a ceiling temperature of 70 F, floor and wall temperatures of 80 F, and 5 air changes per hour at 60 F. A 5 F deg increase or decrease in the ceiling temperature will cause a decrease or increase of approximately 2 percent respectively in the low temperature radiation exchange between the luminaires and the ceiling.

Example No. 3: **Given:** *room air temperature = 75 F; panel surface temperature = 70 F; heat gain from walls and floor = 12 Btu per (hr) (sq ft of panel area); load from direct fluorescent lights = 15 Btu per (hr) (sq ft of panel area).*

To find: (1) *heat removed by the cooled panel;* (2) *amount of 60 F air required to maintain the required conditions; and* (3) *amount of heat removed by the conditioned air.*

TABLE 4—APPROXIMATE DISTRIBUTION OF LIGHTING LOAD FOR DETERMINING HEAT PICKUP BY PANEL AND CONDITIONED AIR, PERCENT

TYPE OF LIGHTING	CONVECTION[a]	RADIATION TO PANEL[b]	RADIATION TO OTHER SURFACES[c]
Direct	60	16	24
Indirect	70	18	12

[a]Values from Table 2; [b]Sum of high- and low-temperature radiation from Table 3; [c]Obtained by subtracting sum of convection and radiation percentages from 100.

Solution: Using the values from Table 4, the following lighting load components may be determined: Convected heat released = 0.6 × 15 = 9 Btu per (hr) (sq ft); Radiation to panel = 0.16 × 15 = 2.4 Btu per (hr) (sq ft); Radiation to other surfaces = 0.24 × 15 = 3.6 Btu per (hr) (sq ft); The sum of the heat gain from the walls and floor, and the radiation from lights to other surfaces = 12 + 3.6 = 15.6 Btu per (hr) (sq ft); The difference between the room air and the ceiling panel temperature is 5 F deg.

Follow along the $t_a - t_p = 5$ curve on Fig. 3 until the sum of the heat pickup by the panel, as read at the left, and the heat pickup by the air, as read at the bottom, is 15.6. This condition is satisfied at an air supply rate of 0.29 cfm per sq ft, and the heat removal by the panel and the air are 10.9 and 4.7 Btu per (hr) (sq ft) respectively.

The air supply must be increased sufficiently to remove the additional 9 Btu per (hr) (sq ft) of convected heat from the lights. Therefore, the total required heat removal by the air must be 9 + 4.7 = 13.7 Btu per (hr) (sq ft), and from Fig. 3, this will require 0.85 cfm per sq ft of floor area.

The total heat removal by the panel will be the 10.9 Btu as just determined from Fig. 3 plus 2.4 Btu direct radiation to the ceiling, or 13.3 Btu per (hr) (sq ft) of panel area.

ROOM-AIR TEMPERATURE GRADIENTS

The room-air temperature gradient curves shown in Fig. 11 were plotted from temperatures taken at the center of the room with No. 36 copper-constantan butt-soldered thermocouples. These were not appreciably affected by radiation, as verified by comparison with an aspirated thermocouple.

With the room-surface temperatures held constant, the room-air temperatures increased as the lighting load increased. The increase near the floor and ceiling was particularly pronounced. Variations in room-air temperature caused by internal convection heaters were about the same as those shown in Fig. 11 resulting from fluorescent lighting.

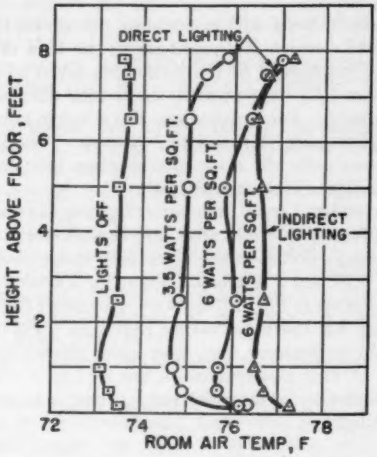


FIG. 11—EFFECT OF LIGHTING LOAD ON ROOM-AIR TEMPERATURE GRADIENTS; (CEILING 70 F, AUST 80 F, CONDITIONED AIR 0.8 CFM PER SQ FT AT 60 F, 10-FT CEILING HEIGHT)

It will be noted that Fig. 11 indicates a higher room-air temperature for indirect than for direct lighting. This is in agreement with the data plotted in Figs. 7, 8, and 9.

REMOVAL OF HEAT FROM OCCUPANTS

One test was made with 8 people in the room, to study the effect of the heat load thus provided, on the heat pickup by the panel. Another test was made with an empty room having identical surface temperatures. The room air temperature was somewhat higher in the test with the occupants because of the heat given up by them. A comparison of results showed that the cooled ceiling-panel picked up approximately 25 percent of the sensible heat output of the occupants. The sensible heat output of the occupants was obtained by an energy balance on the room, and amounted to an average of 229 Btu per (hr) (person). The combined sensible and latent heat loss per person was 438 Btu per (hr).

Additional research must be done before definite conclusions can be drawn on the subject of human occupancy *vs.* panel heat pickup.

Discussion

Much of the research which has been published on the thermal performance of lighting systems in conjunction with air conditioning has dealt with incandescent lighting.

The pioneering work of Leopold[12] introduced the concept of an independent-radiation-transfer, (IRT) which is equivalent to the sum of the low- and high-temperature-radiation transferred to the ceiling, as given in Table 3. Leopold's values of IRT are 1.5 and 3.1 percent for 6 single-tube fluorescent lights, and 12.5 percent for 4-tube fixtures. For comparison, the average values from Table 3 for direct and indirect lighting are 15.6 and 18.3 percent. The data for both studies were obtained with essentially the same room-surface temperatures, but with different air introduction and different fixtures.

The radiant energy emitted from fluorescent lighting is essentially in the visible range of wavelengths (0.4 to 0.75 microns), and in a second range of 2 microns and longer. Very little energy occurs in the ultraviolet range (below 0.4 microns) and in the band between 0.75 and 2 microns. Paint for a cooled ceiling-panel should therefore be a good reflector in the visible range to improve illumination, and should have high absorptivity for low-temperature radiation. Fortunately, most paints which have high light reflectance also have good absorption characteristics for longer wavelengths. A high absorptance in the 0.75 to 2 micron range would be helpful for solar radiation or for incandescent lighting, but would have little value with fluorescent lighting.

Conclusions

The test data and results presented here were obtained in the Environment Laboratory under the specified test conditions. Their application to other spaces at other conditions has not been verified, and the reader is cautioned against their indiscriminate use. However, it is believed that the trends here indicated may serve as a guide in predicting the performance of panel cooling installation.

1. The heat pickup by a cooled ceiling-panel and conditioned air may be summarized as follows: (*a*) the sum of normal heat gain through the room surfaces plus the radiation from lights which falls on the walls and floor is removed in part by the panel and, in part, by the conditioned air, and the division may be determined by Fig. 3, or by Equation 3 of the paper; (*b*) the convection heat gain from the lights is removed almost entirely by the conditioned air system and may be added to the convected load as determined in (*a*); (*c*) the radiation from the lights to the panel can be added to the panel pickup as determined in (*a*).

2. For the particular fixtures used in this study, approximately 10 percent of the energy supplied to the direct lighting system was radiated in the visible wavelengths, 32 percent was emitted as long-wave radiation and 58 percent was transferred to the room air by convection. For indirect lighting, these values were 10 percent visible radiation, 21 percent long-wave radiation, and 69 percent convection.

3. Approximately 16 percent of the energy input to the direct fluorescent lighting fixtures was radiated to the cooled ceiling-panel, and 24 percent to the walls and floor. For indirect lighting, these changed to 18 and 12 percent respectively.

4. More than 20 percent of the energy supplied to the lighting system can be removed

by a cooled ceiling-panel. This includes the heat radiated directly to the ceiling, and that which is reradiated from the other room surfaces.

REFERENCES

1. ASHVE Research Report No. 1444—The ASHVE Environment Laboratory, by Cyril Tasker, C. M. Humphreys, G. V. Parmelee and L. F. Schutrum (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 139).

2. ASHAE Research Report No. 1527—Preliminary Studies of Heat Removal by a Cooled Ceiling Panel, by L. F. Schutrum, John Vouris and T. C. Min (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 95).

3. ASHVE Research Report No. 1453—A Low-Inertia, Low-Resistance Heat Flow Meter, by R. G. Huebscher, L. F. Schutrum, G. V. Parmelee (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 275).

4. ASHAE Research Report No. 1576—Natural Convection and Radiation in a Panel-Heated Room, by T. C. Min, L. F. Schutrum, G. V. Parmelee and J. D. Vouris (ASHAE TRANSACTIONS, Vol. 62, 1956, 337).

5. Reflector and Lighting Equipment Manufacturers Standards Institute, Inc., Third Revised Edition, 1953.

6. General Electric Lamp Bulletin LDI, by C. E. Weitz (Application Engineering Dept., Lamp. Div., Cleveland, Ohio, Oct. 1950, p. 61 & 64.)

7. ASHVE Research Report No. 1317—Influence of Gaseous Radiation in Panel Heating, by F. W. Hutchinson (ASHVE TRANSACTIONS, Vol. 53, 1947, p. 285).

8. *Measurement of Radiant Energy*, by W. E. Forsythes (McGraw-Hill Book Co., Inc. New York, First Edition, 1937, p. 181).

9. Energy Distribution Curve Furnished by Lamp Department of General Electric Co.

10. Relation of Lighting to the Cooling Load, Report of Joint *IES*-ASHAE Committee (*Heating Piping & Air Conditioning*, Sept. 1956, p. 125).

11. ASHVE Research Report No. 1454—Effectiveness and Temperature Requirements for Cooling Panels Removing Internal Radiation, by Merl Baker (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 287).

12. The Mechanism of Heat-Transfer-Panel Cooling, Heat Storage, Part I, Charles S. Leopold (*Refrigeration Engineering*, July, 1947, p. 1).

APPENDIX

CONVECTION TRANSFER TO A COOLED CEILING PANEL

The convective heat transfer to the ceiling was obtained by subtracting from the measured total heat flow into the ceiling surface, the calculated radiation received from the walls and floor of the room. The convective heat transfer (q_c) was then expressed in the conventional form of equation:

$$q_c = h_e(t_e - t_p) \quad \text{(A-1)}$$

where

q_c = convective heat transfer, Btu per (hr) (sq ft).
h_e = equivalent coefficient of heat transfer, Btu per (hr) (sq ft) (F deg) based on $(t_e - t_p)$.
t_e = equivalent film temperature, Fahrenheit.
t_p = ceiling panel surface temperature, Fahrenheit.

The equation for the equivalent film temperature is as follows:

$$t_e = 2.349\ \text{AUST}^{0.58} t_{oa}{}^{0.27} W^{-0.0876} \quad \text{(A-2)}$$

where

AUST = area weighted average surface temperature of the walls and floor, Fahrenheit.
t_{ca} = conditioned-air temperature, Fahrenheit.
W = conditioned-air supply rate, pounds per hour.

Limits: where $t_p = t_{ca}$, $W = 400$ to 2000; $t_p > t_{ca}$, $W = 400$ to 1000.

The equation was evaluated for the specific condition wherein q_e = zero. Thus in Equation A-1, $h_e(t_e - t_p)$ = zero, and therefore, $t_e = t_p$. Experimentally, the variables in Equation A-2 were varied separately so that q_e would equal zero and the equivalent film temperature could be taken equal to the ceiling surface temperature. The exponents of AUST, t_{ca}, and W were then determined by simultaneous equations.

The coefficient of convective heat transfer was equated from Equation A-1.

$$h_e = \frac{q_e}{t_e - t_p} = 0.534\ \mathrm{AUST}^{-.044} t_{ca} W^{-0.24} \quad \ldots\ldots\ldots \quad \text{(A-3)}$$

The limitations of Equation A-3 are the same as for Equation A-2.

MEASUREMENT OF LOW TEMPERATURE RADIATION FROM FLUORESCENT LIGHTING FIXTURES

In order to determine the distribution of the low temperature radiation from direct lighting, one fixture was suspended in the center of the test room. After it had reached operating temperature, the electrical power was disconnected, and the radiation (other than visible) was measured by means of the Korsgaard Radiometer*. This radiometer has a flat surface receiver and thus it exchanged radiation with the room surfaces as well as with the light fixture. Consequently, all room surfaces were controlled at 75 F, and since the emitting surface was small compared with the surrounding room surface area, the surfaces were, in effect, a black body receiver. Radiation measurements were made about the light fixture in 3 vertical planes, normal to the tube axis, parallel to the axis, and at 45 degrees from the axis. The low-temperature-radiation was greatest below the fixture because the fluorescent tubes were the warmest of the exposed surfaces. The ballasts were enclosed in the channel at the top of the fixture, and considerable radiation was directed upwards from the back of the fixture. In contrast, light energy distribution was all directed downward.

The total measured low-temperature radiation from the single fixture amounted to 145 Btu per hr. Assuming that the distribution of low-temperature radiation for each of the 13-light fixtures, when operating together, was the same as for the single fixture tested, the amount of energy intercepted by the ceiling, floor and walls was determined. Interchange factors were calculated for this radiation exchange between the 13-light fixtures and each of the room surfaces. The energy intercepted by each of the room surfaces from the top, sides and bottom of the luminaires was then related to the temperature difference between that surface, and the fixture.

DISCUSSION

WALTER STURROCK†, Cleveland, Ohio, (WRITTEN): From Table 2 it is noted that for direct lighting 41.5 percent of the total energy is in radiant form and from Table 3 that 15.6 percent of the total Btu per hour are absorbed by the ceiling panel at 70 F. This appears to be true for all tests with 5 air changes per hour regardless of the supply

---

*A New Radiometer Measuring Directional Mean Radiant Temperatures, by Vagn Korsgaard (Heating Piping & Air Conditioning, July, 1944, p. 129).

†Large Lamp Dept., General Electric Co.

air temperature. For the indirect lighting the data show 31 percent of the energy in radiant form with 18.3 percent absorbed by the ceiling. This again was true regardless of the supply air temperature.

For comparison with the foregoing I would like to record a few tests with 10 air changes per hour. Although sufficient data were not obtained for conclusive results from the 70 F ceiling they indicated the following:

With direct lighting 12.1 percent of the total energy was absorbed by the ceiling. In this case the 10 air changes per hour apparently kept the lighting unit cooler than that with 5 air changes and therefore there was less low temperature radiation. For indirect lighting the ceiling absorbed 16.5 percent.

As one might expect the data all indicate that ceiling panels for removing heat are considerably more effective for indirect lighting systems than for direct. In fact, the foregoing data show the panel ceiling to be 19 percent better for indirect lighting than for the direct when 5 air changes per hour were used and 36 percent better for 10 air changes per hour.

F. J. LINSENMEYER, Detroit, Mich. (WRITTEN): The parameters chosen for the variables in testing are such as to make the derived units the ones desired to be kept constant. Room temperature in air conditioning is now kept constant at the design condition of 75 to 80 F (as selected by the engineer), and the thermostat setting is changed only to suit occupancy. Further, the air volume supplied to the space is kept constant. By thus maintaining air supply quantity, temperature and humidity constant over the range of space loads from maximum to about ½, the water temperature (or quantity) supplied to the radiant ceiling may be thermostatically varied to suit the load. Under these conditions, the potential latent load pickup of the air is not impaired throughout the load range.

I should like to suggest to the authors that they substitute, in *Examples* 1 and 2, air volume changes of 5 instead of 9/10 and a space cooling load corresponding to 40 or 50 instead of 12 Btu per (hr) (sq ft). An AUST higher than 85 F would be required to simulate such conditions.*

The paper presents data showing the effect of sweeping a cool ceiling with air at or below the panel temperature. Air conditioning is concerned with space loads from the maximum for cooling, through the intermediate season of about equal duration on into the maximum of the heating season. By maintaining air temperatures approximately constant and supplied at space temperature during the winter, the radiant ceiling is swept by air cooler than itself and the convective output is materially increased.

I would like to repeat how much the information contained in this paper is appreciated and look forward to the continued use of the environmental room for comfort studies under radiant panels.

W. F. SPIEGEL, Philadelphia, Pa., (WRITTEN): I have reviewed the paper and wish to congratulate the authors for their work resulting in a simplified approach to this problem through the extensive project at the Laboratory.

Unfortunately, I have some difficulty in understanding parts of the paper and I would like to pass these comments along, under the assumption that others may have had the same problem.

The justification of evaluating heat transfer coefficients at zero transfer is not at all obvious. In previous papers, of Wilkes for example, it appears that there is considerable change of coefficients in the region of small temperature difference.

I would also suggest better labels for the ordinates of Figs. 3, 7, 8, 9 and 10 such as *Panel Pick-Up by Convection and Room Surface Radiation* and *Total Panel Pick-Up*.

---

*If Fig. 3 is used, at 70 F panel, 85 AUST, 60 F supply air, 7½ air changes, 80 F room air, the panel pickup is 29 and the air is 22 for a total of 51 Btu per (hr) (sq ft)—a value more realistic than the one used in *Example* 1 of the paper.

I cannot quite understand the reason why the fictitious use of Fig. 3 should give proper answers. According to the text, this chart is based on equations which take account of actual air velocities at the film at the ceiling. The example then proceeds to evaluate the panel pick-up for 0.085 cfm per sq ft at the 0.29 abscissa. I feel that this point should be clarified.

Low temperature radiation is treated as an independent variable. The fact that it is large, gives added significance to this quantity and it appears almost imperative to determine its variation with ceiling panel temperature and AUST.

Fluorescent tubes operate at approximately 90 F surface temperature in panel-air systems. The average temperature of the fixture is also probably close to this value. It would then make a considerable difference whether the receiver is 75 F, as in the test; 80 F as is the reported AUST; 73 F, which is more nearly the actual AUST in the multi-story panel cooled building; or 65 F which is more nearly the actual panel surface temperature in an extruded panel. If may be recalled that Leopold's work did use a model where the panel temperatures, wall surfaces, and floor temperatures simulated a section of multistory panel cooled structure.

As a minimum, I believe this evaluation should be pointed out and the dissimilarity between this test and multi-story buildings indicated.

The last statement of the paper is not clear. My interpretation is that 20 percent of the lighting input energy is removed from the panel by *radiation* at AUST of 80 F with a 70 F panel; additional amounts are absorbed by convection, depending on how the balance sheet is presented.

JOHN EVERETTS, JR., Philadelphia, Pa. (WRITTEN): If it is possible, I believe an appendix attached to the paper, giving the actual floor, wall and panel temperatures, may help to clarify the problem and also permit other calculations to indicate the order of magnitude of variations which may be expected with changes of room temperature, panel temperatures and AUST.

I appreciate this information is presented in curves but I think it is also well to have calculated data available.

JOHN EVERETTS, JR., Philadelphia, Pa.: There is one other point about which I would like to ask Mr. Schutrum and that is in reference to the work Leopold did in connection with the four-tube fixtures. There appears to be quite a difference in results between Leopold's data and the Laboratory data, which would be accounted for by a difference in the type of heat absorbing paint used in the two experiments.

R. W. MCKINLEY, Pittsburgh, Pennsylvania: I would like to confront the authors from two different points of view. In the paper it is mentioned that a primary interest of engineers is how much of the heat the ceiling panel may be able to pick up. It happens that I am interested in the lighting aspects of engineering and one of the points that this type of study makes is that there is a very close relationship between the electrical engineer, those who are concerned with solar energy transfer into spaces and also the air-conditioning engineer referred to in the paper.

What I want to point out is that some of these data can be used in a different way. We all know that electric radiant heat is already built in most spaces in the form of high capacity electric lighting systems. I believe that we might devote some direct attention to the specific problem of how to better utilize lighting systems as heating systems.

The same data raise the question: Where should the balance line be drawn between lighting and thermal comfort? It seems to me that the objective of all engineers should be to provide a desirable overall environment for the room. Thoughtful compromise between the specialty areas is becoming increasingly important.

W. C. KADOW, Chicago, Illinois: The authors should be complimented on the wonderful work they have done. I would like to cite, for just a minute, work that was

done in our own office in Chicago relative to recessed fixtures. The limited data we came up with while working with General Electric personnel, checking ballast temperatures in a radiant ceiling panel, indicated normal ballast operating temperatures with high water temperatures. Our findings concur with Mr. McKinley's remark that lighting is very important and a serious consideration in the design of building cooling systems and with radiant panels.

B. R. SMALL, Pittsburgh, Pa.: I am happy to work in a building which is panel heated and cooled. In conventionally air-conditioned buildings, the lights heat the ceiling and when the air-conditioning system is shut down, the stored heat is released and many of you know how warm an office building is the next morning.

In the Alcoa Building, there seems to be practically no heat accumulation from the lights. In the summer, they shut off the refrigeration and outside air at 6:00 o'clock yet one can work in the evening and the cool temperature continues right on.

I would like to inquire whether in the Cleveland Research Laboratory tests any of the wall panels were permitted to assume a neutral temperature with no water flow? If so, were there any readings to show a lower wall temperature than the air itself due to the cooling effect of the ceiling? My company built a test room and Mr. Gordon was one of the visitors in 1951. In those tests, we held 80 F room air with about a 65 F ceiling. Some of the wall temperatures were as much as 2½ deg below the room air temperature, thus indicating a worthwhile secondary cooling benefit to the occupants.

In calculating summer cooling loads, I believe most people agree that about 60 to 80 percent of the total room sensible heat load does come from lights and solar heat. Consequently a ceiling panel is very effective under this condition. I remember Professor F. W. Hutchinson commenting on the Leopold panel room that was developed in 1947 and saying that the fascinating point about ceiling panel cooling is that so much of the radiant energy is trapped and removed by the ceiling before it ever becomes sensible heat and before it enters the room.

AUTHORS' CLOSURE (Mr. Min): The authors agree with Mr. Linsenmeyer that an air change rate of about 4 changes per hour would be more realistic than 0.9 changes per hour as used in *Example 1*. The ceiling panel pick-up with 4 air changes per hour would be approximately 12.5 Btu per (hr) (sq ft) as for the conditions of *Example 1*, and including the heat removed by the cool air, the total heat removed per sq ft of floor area would be about 21 Btu per (hr) (sq ft). For the space loading of 40 or 50 Btu per (hr) (sq ft) suggested by Mr. Linsenmeyer, the surface temperature of the ceiling panel would have to be lowered and the air quantity increased according to Fig. 3 of the paper.

The draft of this paper originally included an acknowledgement of the assistance of Walter Sturrock, but it was inadvertantly omitted from the final paper. The authors would like to take this opportunity to thank Mr. Sturrock for his technical and material assistance and his service in these studies.

In his discussion, Mr. Sturrock compared the energy absorbed by the ceiling for direct and indirect lighting, and for 5 and 10 air changes per hour. These percentages include only the radiation components and are based on the total energy dissipated by the lighting.

Mr. Everetts pointed out that there appears to be a difference in results between Leopold's data for 4-tube fluorescent fixtures and the Laboratory data. Furthermore, it is suggested that the heat-absorbing paint used by Leopold may be the reason for these differences. The values of radiation transfer to the ceiling for 4-tube fixtures in the series of tests by Leopold are 12.5 percent for regular paint, and for heat-absorbing paint they are 17.8, 19.6, and 24.5 percent. As pointed out in the discussion of this paper, it is believed that heat-absorbing paint would be helpful in absorbing solar radiation or for incandescent lighting, but would have little value for fluorescent lighting; and consequently, the differences in the percentages just stated could be attributed to experimental scattering. Assuming this to be true, the average of the 4 percentages

is about 18.6 percent which is in good agreement with the range of values of radiation absorbed by the panel, 8.1 to 19.7 percent as given in Table 3 of the paper.

The question of evaluating the convection heat transfer coefficients has been raised by Mr. Spiegel. The authors certainly appreciate his interest and constructive comments.

A description of the method used for determining the convention coefficients is in the Appendix. It should be added that the zero convective heat transfer concept is used in determining the boundary conditions of the equivalent film temperatures.

The convective heat transfer for a ceiling panel in a ceiling-panel-cooled room with introduction of cooled air through the ceiling diffuser is rather complex. It is influenced by many factors such as the AUST, the cooled air temperature, the cooled air supply rate and ceiling panel temperature. The fundamental equation of convective heat transfer usually employed is the socalled *Newton's cooling law.*

$$q_c = hA(t_s - t_f)$$

where

q_c = convective heat transfer.
h = convective heat transfer coefficient.
A = area of heat transfer surface.
t_s = temperature of surface.
t_f = temperature of the fluid.

However it is rather a definition of h than a law*, and the value of h depends upon the definition of t_f or upon where the temperature of the fluid t_f is measured. Conventionally t_f is the bulk temperature or the temperature of the main body of the fluid, *i.e.* the temperature of the fluid just outside the boundary layer.*

In a ceiling-panel-cooled room with the introduction of cooled air through the ceiling diffuser, the room air is at one temperature, and the cooled air or conditioned air introduced through ceiling diffuser is at another temperature. This air stream after leaving the diffuser will be distributing near by the ceiling. The temperature of the cooler air is at t_m measured at the throat of the diffuser. After leaving the diffuser, the temperature of this air stream, however, varies with horizontal distance from the diffuser as well as the vertical distance from the ceiling panel.

To find these air temperatures at various vertical distances from the ceiling and horizontal distances from the diffuser, and to determine where to measure the characteristic air temperature for the convection heat transfer for the ceiling is beyond the scope of the present investigation. In this study, the chief concern was the thermal radiation to the ceiling from other room surfaces, low temperature radiation to the ceiling from luminaires and fixtures, short wave radiation to the ceiling from lights, (direct lighting and indirect lighting), heat pick-up by cooled air or conditioned air, in addition to the convection from air to ceiling.

The convective heat pick-up for a 70 F ceiling panel for example at a 85 F AUST, 60 F cooled air temperature (measured at the throat of diffuser) and cooled air rate of 4 air changes per hour as shown in Fig. 2 in the paper is 3.25 Btu per (hr) (sq ft). When the cooled air rate increases to 18 air changes per hour, the ceiling panel will give away convective heat of 0.7 Btu per (hr) (sq ft). In both cases, the room air temperatures are greater than ceiling panel surface temperatures, and the cooled air temperatures measured at the throat of the diffuser are smaller than the ceiling panel surface temperatures; yet in the first case, the ceiling picks up the convective heat, and in the second case, ceiling gives away the convective heat. To correlate the convection heat transfer with either the temperature difference between ceiling panel and room air or between the ceiling panel and cooled air is improbable. This dilemma is solved

---

**Heat Transfer* by Max Jakob, (John Wiley and Sons, New York, Vol. I, 1949, p. 2 and p. 13).

by the use of the concept of zero heat transfer and equivalent film temperature t_e. The convection heat transfer may then be determined by Equation 2 in the text.

The equivalent film temperature is that fictitious air temperature which in the absence of all other effects, such as the AUST or the temperature and supply rate of the cooled air from the ceiling diffuser, would give the same rate of convective heat transfer from the ceiling for a given ceiling panel temperature as would exist under the actual combination of AUST, the temperature and supply rate of the cooled air from the ceiling diffuser. In other words, under the combined effect of AUST, the temperature and the supply rate of cooled air from the ceiling diffuser, the temperature of the air stream "*sweeping*" near by the ceiling panel varies from one place to another but could be equivalent to the air having a certain uniform temperature, just outside the boundary layer, which is defined as equivalent film temperature and which would give the same convective heat transfer.

The determination of the equivalent film temperatures is as follows:

From a group of tests wherein the AUST, the temperatures of cooled air (t_{ca}), the temperatures of ceiling panel (t_p) are the same but the cooled air mass rates through the diffuser (w) are different, we plot the convective heat transfer for the ceiling (q_c) against W, and find at W_1 the q_{c1} is equal to zero. By the concept of zero heat transfer and of the equivalent film temperature stated, t_e, is equal to t_p, at W_1, AUST$_1$, and t_{ca1}. Assuming a power function, t_e may be expressed as

$$t_{e1} = t_{p1} = a(\text{AUST}_1)^b(t_{ca1})^c(W_1)^d$$

where a is an experimental constant and b, c, d, the experimental exponents. Likewise, plot q_c against t_{ca} for the test with same AUST$_2$, W_2, t_{p2}; and when $q_{c2} = 0$ at t_{ca2}, establish another boundary condition of t_e, *i.e.*

$$t_{e2} = t_{p2} = a(\text{AUST}_2)^b(t_{ca2})^c(W_2)^d$$

and so on. The constant a, and exponents b, c, d were determined by the simultaneous equations or by the relaxation method, and were checked with other tests not used in evaluating the boundary conditions of t_e. The convection heat transfer coefficient h_c based on the temperature difference of $(t_e - t_p)$ is then correlated by a power function.

$$h_c = q_c/(t_e - t_p) = e(\text{AUST})^f(t_{ca})^g(W)^j$$

For any given test, say, AUST$_6$, T_{ca6}, W_6, t_{p6}, q_{c6}, establish

$$h_{c6} = q_{c6}/(t_{e6} - t_{p6}) = e(\text{AUST}_6)^f(t_{ca6})^g(W_6)^j$$

where

$$t_{e6} = a(\text{AUST}_6)^b(t_{ca6})^c(W_6)^d$$

The constant e and exponents f, g, j again can be determined by simultaneous equations from the boundary conditions of h_c.

It is understood that the values of the constants a, e and exponents b, c, d, f, g, j are valid only for a given type of diffuser at a given orientation. But the concept and technique of correlation could be applied to other types and orientations. Incidentally after this study and correlation were made, the authors learned that the similar concept has been employed in heat transfer of high velocity flow.*

The authors are grateful to see Mr. Spiegel is interested in this complicated convection mechanism, and would like to see that a particular study on the temperature distribution of the cooled air through different types of ceiling diffusers at various orienta-

---

***Heat Transmission* by W. H. McAdams, (McGraw-Hill Book Co., New York Third edition, 1954, p. 313).

tion and on the corresponding convective heat transfer for the ceiling panel in a ceiling-panel-cooled room be encouraged.

Fig. 3 as questioned by Mr. Spiegel was determined, in part, by use of these empirical convection equations, by making a heat balance on the room. Average curves are drawn for constant room-air to panel-surface temperature differences. Fig. 3 applies approximately to the lighted room condition with the following considerations:

(a) The convected heat from the lighting is assumed to be picked-up by additional conditioned air supplied to maintain constant room conditions which are similar to that for internal convective heat loads discussed in the paper. This is approximate but was found to be essentially true for the test conditions.

(*b*) Radiation from the lighting which is absorbed by the ceiling panel is taken as a constant value of 16 percent for direct lighting and 18 percent for indirect. However, low temperature radiation dissipated by the lighting and absorbed by the panel is not a constant value; and consequently, an empirical equation was developed to express this heat transfer as a function of the AUST, panel surface temperature, conditioned-air temperature and rate, and the energy input to the lighting system. The variation of this low-temperature radiation absorbed by the ceiling was judged not to be of sufficient magnitude to warrant this refinement for the lighting intensities used in the tests (6 watt per square foot of floor area).

(c) The apparent conduction load through the walls and floor as normally calculated for heat gain estimates was assumed to equal the net heat transfer from these surfaces as observed in the experiments. The net heat transfer from the wall would be the sum of the convection transfer to the room air, the radiation transfer to the panel and other room surfaces, and the radiation transfer from the lighting system to the wall. The conduction load is normally calculated without considering any radiant internal heat sources and no attempt was made to refine this calculation.

The bulb wall temperatures of the fluorescent lights as measured in these tests are given in a paper titled *A Study of the Extraction of Heat from Fluorescent Luminaires in Cooled Rooms** and are considerably higher than the 90 F suggested by Mr. Spiegel.

In the aforementioned paper, Mr Sturrock reported some information obtained from the same experiments as those reported in the paper under discussion today. A close comparison turns up an apparent discrepancy which can be explained. The emphasis in Mr. Sturrock's paper was placed on the lighting rather than on the viewpoint of the air-conditioning engineer, and, as a result, corrections to the light output (illumination) were made for the variation in bulb wall temperatures. In the present paper this difference was neglected in that it amounted to one or two percentage points and would not be significant.

Unfortunately, the authors have no test data with a neutral wall condition to compare with Mr. Small's experiment of 2½ F deg depression of wall temperature below room air temperature.

**Illuminating Engineering*, November 1957, by Walter Sturrock and Lester F. Schutrum.

1632

No. 1632

RADIANT DRAFTS FROM COLD CEILINGS†

By HANS E. RONGE*, M.D., AND BORJE E. LOFSTEDT*, M.B., UPPSALA, SWEDEN

THE LOSS of *dry* heat from the human body to the surrounding room is due to radiation and convection to surfaces or to air at lower temperatures than the body surfaces. It was not until the introduction of partitional calorimetry[1, 2] that it became possible to study the mutual significance of these 2 cooling processes. These investigations have since been widened considerably[3, 4, 5]. In Denmark, the problem has been dealt with by Pedersen[6, 7] and by Nielsen and Pedersen[8]. As variables such as velocity of air, temperature of skin and clothing surface, and the magnitude of the effective radiation or convection surface are included, the mutual relation between radiation and convection heat losses cannot be stated by a constant. For normal indoor conditions with the same air and radiation temperatures and the subject sitting or standing still and with an air velocity of 5-10 cm per sec it would seem that the convection losses preponderate to some extent over the radiation losses, partly due to the effective convection surface being larger than the effective radiating surface of the human body. Thus for standard conditions[1] with air velocity of 7.6 cm per sec the total radiation constant (with the effective body surface taken into account) is responsible for 48 percent of the total dry heat loss and the convection constant for 52 percent. Similar results, though with rather greater preponderance for the convection, were found by Nielsen and Pedersen[8]. These investigations apply to loss of heat from the *entire* human body.

A certain interest, not least from the practical point of view, lies also in investigations concerning convection and radiation losses from small, local surface areas of the human body. In the present investigation the effect of a cold ceiling on the temperature of the shoulders and neck and on the heat balance of the human body was studied. For certain premises with high rate of heat production, cooling by means of cold ceilings constitutes a suitable and practical method for carry-

†The investigation was performed at the Climatic Physiological Laboratory, Uppsala University, with financial support from the Swedish Building Research Board.
*The Climatic Physiological Laboratory, Uppsala University.
[1]Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, Pa., January, 1958.

ing away surplus heat. It is then of practical interest to know the extent to which the cold ceiling can give rise to cold or draft sensations on the people on the premises and the extent to which this can be compensated by higher air temperatures. In this connection, there may be an influence from the circumstance that in many persons shoulders and neck appear to be particularly sensitive to cooling by draft with subjective disagreeable, sometimes *rheumatic*, reactions.

METHODS AND APPARATUS

Conduct of Experiments: The experiments were carried out in a climatized room, in one part of which a cold ceiling was installed (see Fig. 1). Owing to the small size of the room only one subject could be tested at a time. The subject was first

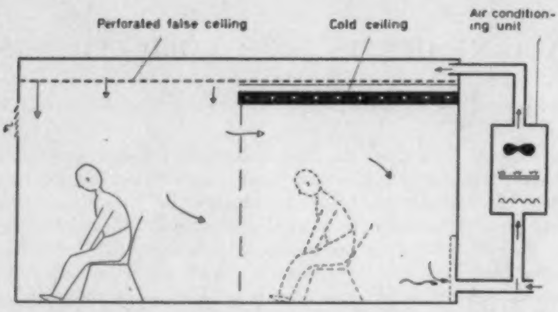


FIG. 1—SKETCH OF THE TEST ARRANGEMENTS

placed in the *reference room* beneath the *warm ceiling*, until temperature equilibrium was established on the skin and particularly on the shoulders, neck and other body surfaces exposed to the ceiling (30 to 45 min), after which he was moved beneath the cold ceiling. New temperature recordings were then made until eventually a new temperature equilibrium was reached (60 to 120 min). Along with the subjective statements, the finger and toe temperatures were used as an objective measure as to whether a general reaction to cold (chill reaction) develops. Of the experiments performed, totalling 20 (see Table 1) 7 comprised experiments with naked upper body and with the body at rest, 6 with one layer of clothes (thick working shirt) over the upper part of the body and with the body at rest, and 7 experiments with 2 to 3 layers of clothes (corresponding to normal working clothes) and light manual work (total energy development 150 kcal per hr = 595 Btu per hr) on an ergometer cycle.

Besides these experiments on a model scale, opportunity was provided for an investigation on large factory premises below ground, where use was made of the ceiling cooling system. In this way a practical check on a random test basis was obtained of the laboratory experimental results.

Experimental Apparatus: The climatized room (Fig. 1) was ventilated by an air-conditioning unit for heating and moistening and blowing in the preconditioned

air through a perforated false ceiling. The air supply of the room was equivalent to 10 air changes per hour. The amounts of fresh and return air could be varied by damper. The relative humidity was maintained at 55-60 percent for all tests

TABLE 1—SURVEY OF TESTS AND TEST CONDITIONS AND SKIN TEMPERATURES AFTER SPENDING 0.5-1 HR. IN REFERENCE ROOM AND BENEATH COLD CEILING

TEST No.	SUBJECT	CLOTHES AND WORK	AIR C	REF. CEILING C	COLD CEILING C	SHOULDERS REF. C	SHOULDERS COLD CEILING C	FINGERS REF. C	FINGERS COLD CEIL. C	REMARKS
1	A	Upper body, naked at rest	18.8	19.2	13.7	31.0	29.9	..	..	Chill reaction
2	Ae	Upper body, naked at rest	19.3	19.7	13.0	31.3	29.7	..	..	Chill reaction
3	L	Upper body, naked at rest	25.0	27.2	16.0	33.3	31.5	34	33[a]	
4	Lt	Upper body, naked at rest	23.2	26.0	..	33.7	..	32	..	Subjectively indiff.
5	Lt	Upper body, naked at rest	26.0	27.0	17.5	33.8	32.5	34	27	
6	A	Upper body, naked at rest	28.5	35.0	18.0	34.4	33.0	35	34	
6x	T	Upper body, naked at rest	18.5	..	13.5	..	29.2	..	26[b]	
7	Lt	One layer clothing, at rest	16.6	16.3	12.5	32.4[c]	31.5[c]	19	17	
8	Lt	One layer clothing, at rest	21.0	23.5	14.5	32.8	31.3	33	23	
9	Lt	One layer clothing, at rest	23.0	24.0	15.5	33.4	32.0	33	27[d]	
10	S	One layer clothing, at rest	23.2	26.0	..	33.5	..	30	..	
11	Lt	One layer clothing, at rest	26.2	31.0	17.0	34.2	32.6	35	33	
12	L	Two layers clothing, at rest	25.0	28.0	16.5	35.0	34.0	33	32	
13	Lt	2 to 3 layers cloth., at work[e]	15.6	16.6	12.0	32.1	31.7	21	20	
14	Lt	2 to 3 layers cloth., at work[e]	17.5	18.0	12.8	32.0	31.3	30	20	
15	Lt	2 to 3 layers cloth., at work[e]	18.0	18.5	12.5	33.4	33.0	28	24[f]	
16	Lt	2 to 3 layers cloth., at work[e]	20.0	21.5	14.0	33.5	33.5	32	32	
17	Lt	2 to 3 layers cloth., at work[e]	22.2	24.4	15.0	33.3	32.9	35	33	
18	Lt	2 to 3 layers cloth., at work[e]	24.0	26.6	16.0	34.4	33.5	33	33	
19	Lt	One layer cloth., at work[g]	19.0	19.5	14.4	33.0	32.2	30	28	

[a]Chill reaction after 2 hr. [b]Falling; 2-hr test. [c]Local collodium irritation. [d]Subjectively indiff. [e]At the rate of 150 kcal per hr. [f]Falling. [g]At the rate of 225 kcal per hr.

on persons. A cold ceiling, consisting of a concrete slab, 2.8 × 2.0 m (9.2 × 6.5 ft), with pipe coils inset, the piping beginning about 0.5 m (19⅝ in.) apart, was set up independently on posts above part of the test room, 2.1 m (7 ft) above the floor. The pipe coils were fed with cold water at the rate of 13 lit per min (3.4 gpm) the temperature on entering being 8.5 C (47.3 F). The subject was placed beneath the cold ceiling on a chair or a saddle seat 0.9 m (3 ft) above the floor so that the

shoulders were exposed to the cold ceiling at the space angle (about 2.2 steradians†) such as would be the case in a workshop with ceiling height of 6 m (20 ft) and width of 11 m (36 ft). A downwardly directed flow of cold air from the cold ceiling could be observed just under the ceiling by the use of titanium tetrachloride fumes. After that, the cold air became entirely mixed with the somewhat rising circulating air. Thus cooling of the subjects due to the downward flow of cold air from the ceiling did not occur to any measurable extent.

Owing to the constant delivery of cold water the underside of the cold ceiling assumed a definite surface temperature for each level of air temperature in the

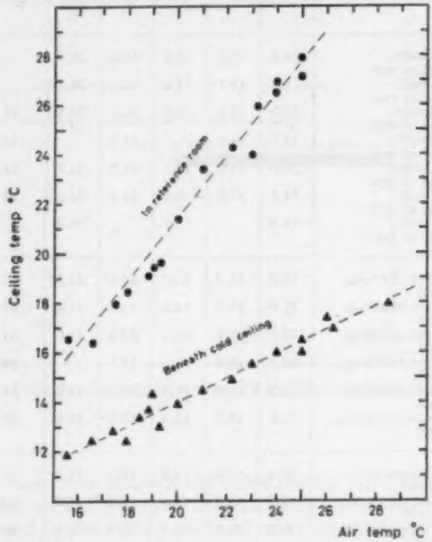


FIG. 2—RELATION BETWEEN CEILING SURFACE TEMPERATURE AND AIR TEMPERATURE PREVAILING IN TEST ROOM

room (see Fig. 2). Thus, with an air temperature of 18 C (65 F) there was a cold ceiling temperature of 13 C (55 F), while the lowest cold ceiling temperature at 28 C (82 F) air temperature was about 18 C (65 F). Condensation occurred on the cold ceiling only if the relative humidity in the premises exceeded 60-65 percent. The vertical temperature gradient beneath the cold ceiling from the floor up to 1 dm (4 in.) below the ceiling surface was comparatively small, less than 2 C deg.

In the reference section of the room, the temperature of the ceiling was equal to

†The solid angle subtended at the center of the sphere by a portion of the surface whose area is equal to the square of the radius of the sphere.

or a little higher than that of the air, especially with high air temperatures (see Fig. 2). In the latter cases, temperature gradients up to 8 C deg were measured between the floor and 1 dm below the ceiling. The air temperature figures which follow were measured in the vicinity of the subject by high-speed recording instruments.

Owing to the efficient air exchange the air temperature beneath the reference ceiling and the cold ceiling was practically identical at each experiment.

The 20 complete tests were performed with 5 separate subjects aged from 20 to 35 years.

Measuring Methods: Measurements of temperature in the air, on skin and on room surfaces were taken chiefly with a thermoelectric instrument having an adjustment time of about 2 sec. In some tests, the thermocouples were fixed to the forehead, the shoulders, the neck, between the shoulder blades, to the chest and to the finger-tips by means of tape, but in such a way that the tape did not cover the soldered tip of the thermocouple. By means of a switch it was possible to read any desired thermocouple on the galvanometer. In one case (test 7) the thermocouples were fixed with collodium, but this was found unsuitable, as a local skin reaction arose accompanied by reddening around the place of application. In other cases, the skin temperature measurements were taken at regular intervals by means of a suitable pencil-shaped applicator.

Besides direct measurements with thermocouples, the surface temperatures of ceilings and clothes were measured by means of a radiation sensitive Moll thermopile which was calibrated and checked by means of a sooted Leslie cube filled with water at known temperature. As the surface temperature of the cold ceiling varied several degrees, according to the location of the measuring point in relation to the underlying cooling coils, the determination of radiation temperature was found to be a more convenient method. For determination of clothing surface temperature the radiation method is decidedly superior to other methods. No correction was made for emission factors of the different surfaces, as both the rough concrete surface and the clothing surfaces may be regarded as having an emission factor very close to 1.0.

Humidity measurements were taken both by calibrated hair hygrometers and by psychrometers.

For the work tests a cycle ergometer with oil pump braking was used to insure a constant and reproducible work intensity. By determining the oxygen consumption by current clinical methods (Krogh's closed method) the load and pedalling speed were so adjusted that the work represented a heat generation of 150 kcal per hr, which corresponds to the heat which a person standing at a workshop bench would generate at light work.

Subjects were asked at regular intervals to state their feeling of warmth or cold according to a special figure scale. As an objective measure of possibly arising general chill reaction the temperatures at the tips of the fingers and toes were mainly used.

RESULTS

As each test produced a record with 100 or so skin temperature figures, which owing to the different conditions of tests cannot be assembled into group or mean figures, presentation of the entire material is difficult. Table 1 gives a summary of test conditions and of equilibrium figures for skin temperatures on the shoulders and at the fingertips (mean values of right and left sides) at the end of the stay

beneath the reference ceiling and the cold ceiling respectively. Changes in finger temperature constitute a criterion for any alterations in the body's total heat balance.

Examples of Recording Curves: Fig. 3 shows the temperatures on one shoulder with the upper part of the body naked during one of the first tests, from which the distinct effect of the cold roof may be seen. Fig. 4 shows the complete temperature record of test No. 3 with air temperature at 25 C (77 F), reference ceiling at 27.2 C (81 F), cold ceiling at 16.0 C (61 F). During the stay in the reference room the heat equilibrium is maintained and is inclined to move towards heat reaction (see

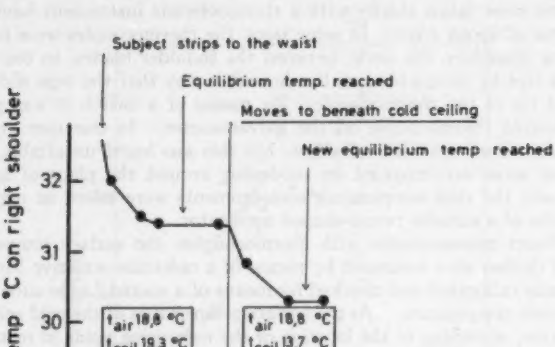


FIG. 3—EXAMPLE OF TEMPERATURE RECORD ON MOVING TO BENEATH COLD CEILING

toe temperature curve). After moving under the cold ceiling there occurs a momentary fall of the temperatures, particularly in the skin surfaces exposed horizontally to the ceiling, and after about 2 hours a general chill reaction is developed.

Upper Body Naked at Rest: This test series (tests 1-6x) was carried out chiefly to find out whether, in general, the cold ceiling has any influence on the temperature and heat balance. The tests showed clearly that such an influence was present. The cooling effect of the ceiling is particularly pronounced on the shoulders, the neck, the upper part of the back (between the shoulder blades), and on the surface of the hair on the crown of the head; but the temperature on the underhair was but little affected or possibly only temporarily. The areas mentioned are evidently most exposed to heat radiation towards the cold ceiling with slightly forward inclined body position.

The cooling action of the ceiling makes itself apparent in a temperature fall on the shoulders of 1.5 to 2.5 deg C, counting from the equilibrium figure in the reference room (see Table 1). During the fall of temperature there constantly arose a clear feeling of chill (*draft*) over the shoulders and neck. If the shoulder temperatures reached figures below about 31.5 C (88.7 F), the chill persisted even after the initial steep fall in temperature. In these cases a general chill reaction soon

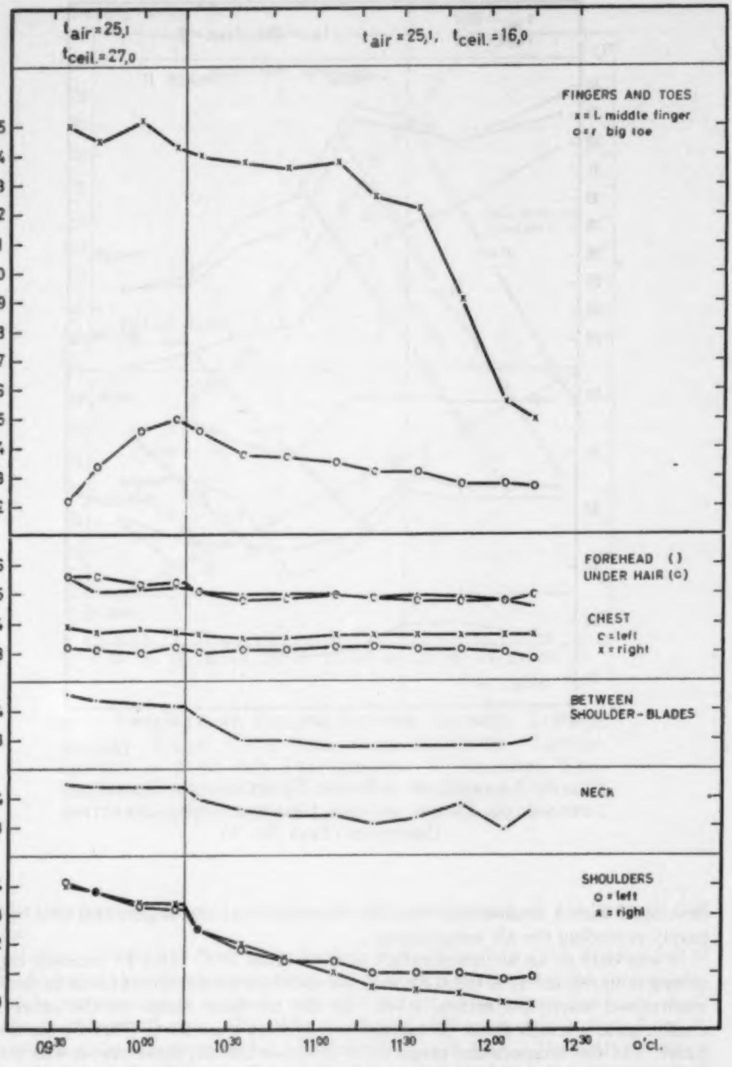


FIG. 4—EXAMPLE OF COMPLETE RECORD FOR A TEST (No. 3). THE MOVE TO BENEATH THE COLD CEILING TOOK PLACE AT 10:15 A.M.

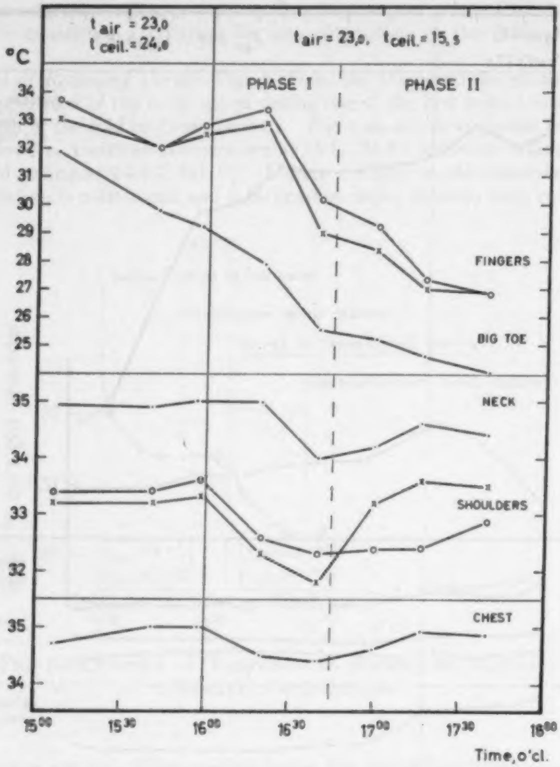


FIG. 5—EXAMPLE OF 2-PHASE TEMPERATURE COURSE ON SURFACE OF TRUNK AS THE GENERAL CHILL REACTION DEVELOPS (TEST NO. 9)

developed with a pronounced drop in temperature at the fingers and toes to levels barely exceeding the air temperature.

It was only at an air temperature of more than 26 C (78.8 F) beneath the cold ceiling temperature 17 C (62.6 F), that the shoulder temperatures could be definitely maintained above the critical level. In the reference room, on the other hand, comfortable warmth could be maintained with at least 23 C (73.4 F) air temperature. In the temperature range of 23-26 C for the air, therefore, it was the cold ceiling, for persons with the upper body naked, that was the weight in the scale which brought the body down to negative heat balance with chill reactions as a result.

In some of the aforementioned tests, it was observed that the temperature fall on the shoulders after moving under the cold ceiling was succeeded by a rising

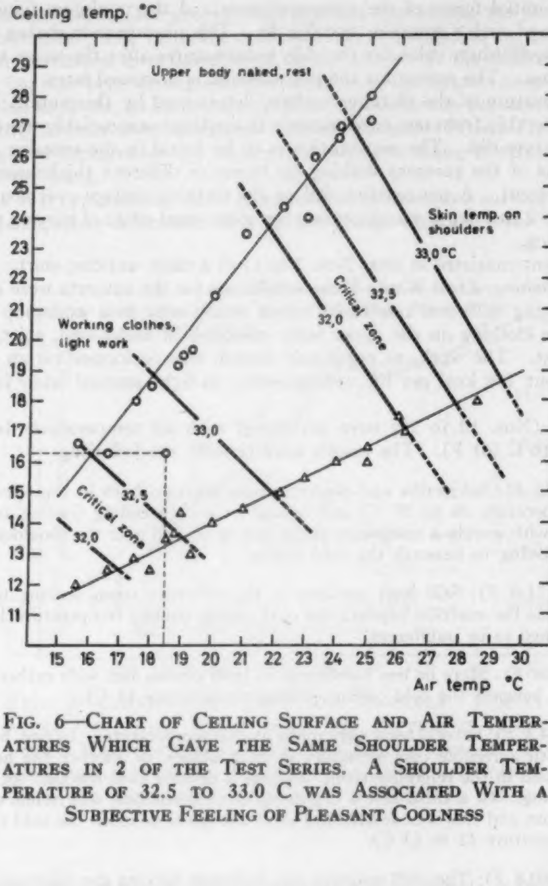


FIG. 6—CHART OF CEILING SURFACE AND AIR TEMPERATURES WHICH GAVE THE SAME SHOULDER TEMPERATURES IN 2 OF THE TEST SERIES. A SHOULDER TEMPERATURE OF 32.5 TO 33.0 C WAS ASSOCIATED WITH A SUBJECTIVE FEELING OF PLEASANT COOLNESS

shoulder temperature in conjunction with the development of the general chill reaction.

One Layer of Clothes, at Rest: Even in these tests (Nos. 7 to 12) a distinct fall of about 1 C deg in shoulder temperatures occurs after moving under the cold ceiling. The fall in temperature is regularly accompanied by a temporary feeling of draft over the shoulders and neck. Persistent feeling of chill arose only in tests 7 and 8, where the shoulder temperature fell to below 31.5 C (88.7 F). The limiting values for threatening chill reaction in the reference room lay at the air temperature 20 C (ceiling temperature 21 C), but the corresponding figures beneath the cold ceiling were 23 to 23.5 and 15.5 to 16.0 C, respectively.

In several of these tests it was observed that the skin temperature on shoulders, under-hair and neck after the initial fall rose somewhat to an equilibrium value

between the initial figure of the reference room and the minimum figure. Fig. 3 is an example of such a reaction (test No. 9). This phenomenon makes it difficult to state an equilibrium value for the skin temperatures after the move to beneath the cold ceiling. The reason for the phenomenon is discussed later.

The temperature of the clothing surface, determined by thermopile, in general varied considerably from one measurement to another—appreciably more than the skin temperature did. The explanation is to be found in the creasing and other displacements of the garment making air layers of different thicknesses between skin and garment. A temperature fall on the clothing surface over the shoulders amounting to 2 to 3 deg C was, however, the most usual effect of moving to beneath the cold ceiling.

The garment consisted in tests Nos. 7 to 11 of a thick working shirt.

Normal Clothing, Light Work: These conditions for the subjects were considered as best agreeing with real conditions which would arise in a workshop with cold ceiling. The clothing on the upper body consisted of under-vest, shirt, and blue overall jacket. The work, as previously stated, was performed on an ergometer cycle at about 150 kcal per hr, corresponding to light manual labor in standing position.

The tests (Nos. 13 to 19) were performed with air temperatures between 24 (75 F) and 16 C (61 F). The results were broadly the following:

At 24 *C* (75 *F*): Subjective and objective heat reaction both in the reference room (ceiling temperature 26 to 27 C) and beneath the cold ceiling (ceiling temperature 15 to 16 C), with merely a temporary slight feeling of chill over the shoulders and neck just after moving to beneath the cold ceiling.

At 22 *C* (71.6 *F*): Still heat reaction in the reference room (ceiling temperature 24 C), whereas the reaction beneath the cold ceiling (ceiling temperature 15 C (59 F) is more inclined to be indifferent.

At 20 *C* (68 *F*): More or less indifferent in both places, but with rather lower skin temperature beneath the cold ceiling (ceiling temperature 14 C).

At 18 *C* (64.4 *F*): Several tests were made at this temperature. In one, heat balance prevailed with agreeable cool sensation in both places, in another the heat balance was maintained in the reference room, whereas a definite chill reaction set in beneath the cold ceiling. In a third test a beginning of chill reaction was noted while in the reference room and this was accelerated after the move beneath the cold ceiling (cold ceiling temperature 12 to 13 C).

At 16 *C* (60.8 *F*): The chill reaction sets in before leaving the reference room..

The temperature drop on the shoulders after moving was in these cases less throughout than in the series described earlier, and amounted to from 0.5 to 1 C deg at the most.

Analysis and Discussion of Results

2-Phase Temperature Reaction: The phenomenon observed in some of the tests with first falling and then rising temperature on shoulders and neck after moving to beneath the cold ceiling, as illustrated by Fig. 5, would appear to be capable of explanation in the following manner. To begin with, the temperature of the trunk surface falls on account of the physically caused, increased cooling and soon there develops a general chill reaction with falling finger and toe temperatures (*phase I*).

The vasoconstriction in the general chill reaction first affects the peripheral ends of the extremities. The resulting decreased blood circulation in the extremities leads to an increase of circulating blood in the trunk. If the local chill effect on the trunk has not been strong enough to produce maximum vasoconstriction in the skin, this leads also to an increase in blood circulation in the skin of the trunk, with resulting increase of temperature. During this *phase II* (see Fig. 5) there are still falling temperatures on the fingers and toes, but rising temperatures on the surfaces of the trunk.

Based on this explanation, this 2-phase course of the trunk surface temperature would be particularly noticeable with moderate chill reaction, when the vasocon-

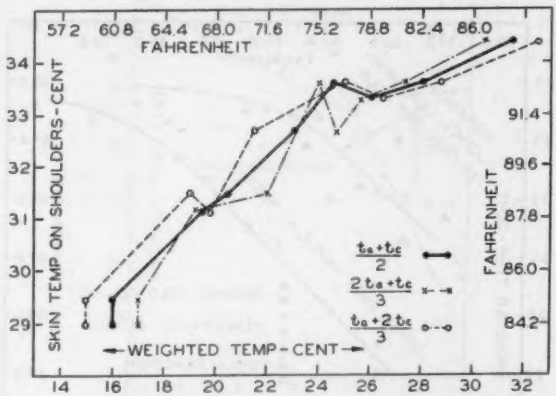


FIG. 7—CORRELATION BETWEEN SHOULDER TEMPERATURE ON NAKED UPPER BODY AND DIFFERENT COMBINATION MEASUREMENTS OF AIR AND CEILING SURFACE TEMPERATURES (t_a AND t_c). THE FLATTEST CURVE IS OBTAINED WITH THE ARITHMETICAL MEAN OF THE AIR AND CEILING SURFACE TEMPERATURES

striction in the skin is chiefly confined to the extremities. This is, indeed, supported by the present test material; it was especially with shoulder temperatures around 32 C (89.6 F) during the first phase that a second phase with rising shoulder temperatures appeared. With greater local temperature fall over the shoulders and stronger general chill reaction the effect was not noticed.

In many of the tests this secondary temperatures increase on the trunk surface was succeeded by a weak feeling of warmth.

Knowledge of this 2-phase reaction is evidently of great significance in all investigations and analyses of local *draft* effects. The difficulty of applying physical cooling equations, at least to isolated parts of the human body, is also illuminated by this.

Air and Ceiling Temperatures in Comfort Chart: In Fig. 6 the results of 2 of the test series (naked upper body at rest, and working clothes with light work) have

been combined to make a *comfort chart* with respect to air and ceiling surface temperatures. The temperature on the shoulders is included in the chart as parameter. These *isotherms* have been so drawn as straight lines that they fit best with all the recorded equilibrium values for shoulder temperature. It appears from the test results that a general chill reaction arose in the subjects when the skin temperature on the shoulders fell below about 32 C (89.6 F); with a shoulder temperature of 31.5 C (88.7 F) or lower the chill reaction set in very rapidly, whereas an equilibrium temperature of 32 C (89.6 F) could be maintained for one or a few hours before the manifest chill reaction arose. A shoulder temperature of 32.5 to 33 C was in all tests associated with a subjective feeling of *pleasantly cool*, while shoulder temperatures of 34 C (93.2 F) or over were associated with subjec-

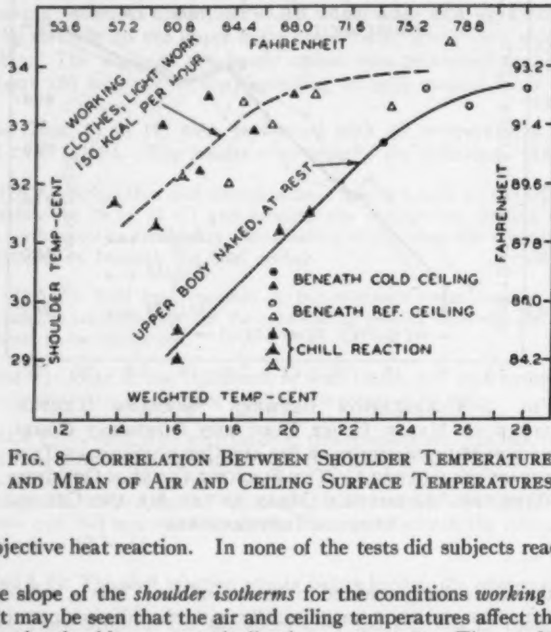


FIG. 8—CORRELATION BETWEEN SHOULDER TEMPERATURE AND MEAN OF AIR AND CEILING SURFACE TEMPERATURES

tive and objective heat reaction. In none of the tests did subjects reach sweating limit.

From the slope of the *shoulder isotherms* for the conditions *working clothes with light work* it may be seen that the air and ceiling temperatures affect the skin temperature on the shoulders to practically the same extent. The resulting or estimated temperature, therefore, is best obtained as the arithmetical mean of the air and ceiling temperatures. On the whole the chart agrees very well with the chart of *resulting temperature* which Pedersen and Nielsen have recently published for the whole of the human body in a room with different air and radiation temperatures.

Combination Measurements of Air and Ceiling Surface Temperatures: In Fig. 7 is shown the relation between the skin temperature of the shoulders on naked upper body at rest and some separate combination measurements of air and ceiling surface temperatures. The simple arithmetical mean between the air and ceiling temperatures, *i.e.* $(t_a + t_c)/2$, gives the flattest curve, as may be seen. The curve

for $(2t_a + t_o)/3$ corresponds to the relation originally given by Pedersen[6] for resulting temperature, while the curve $(t_a + 2t_o)/3$ corresponds to the relation found by Hardy and Du Bois[2]. Recent investigations, both American and Danish, however, have shown that the resulting temperature in general corresponds more to the arithmetical mean of air and radiation temperatures, and this is supported also by this investigation on limited skin surfaces.

In Fig. 8 the shoulder temperatures for the 2 cases, naked upper body at rest and working clothes with work, are shown against the arithmetically estimated

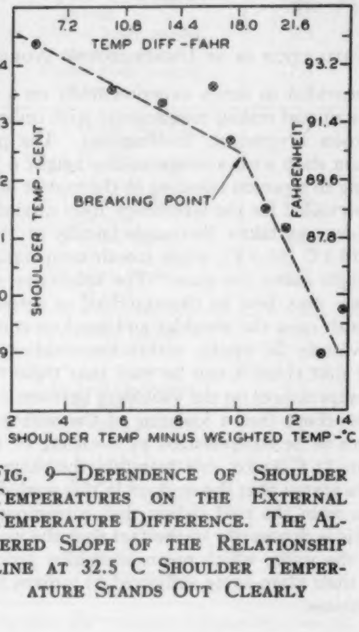


FIG. 9—DEPENDENCE OF SHOULDER TEMPERATURES ON THE EXTERNAL TEMPERATURE DIFFERENCE. THE ALTERED SLOPE OF THE RELATIONSHIP LINE AT 32.5 C SHOULDER TEMPERATURE STANDS OUT CLEARLY

resulting temperature. For shoulder temperatures below about 32.5 C (90.5 F) there prevails, as may be seen, a lineal relation to the weighted temperature. The increased dispersion for tests with working clothes and work would seem mainly to be due to differences in thickness of clothing and insulation from garments.

For the tests with naked upper body, there is given in Fig. 9 the relation between the equilibrium values of the shoulder temperature and the difference prevailing at the same time between shoulder temperature and weighted ceiling-air temperature. More distinctly than previously, it stands out there that the temperature curve for the shoulders changes its inclination for a shoulder temperature of 32 to 32.5 C. This is graphically shown on the figure as a ***breaking point***. The extent to which this point coincides with the local vasoconstriction or constitutes a sign of incipient arterial precooling, according to Bazett[9], it was not possible to decide

with certainty. (It is possible that an arterial precooling mechanism cannot in any case arise on this skin area, owing to the anatomical blood vessel conditions.)

Finally, a comparison has been made in Table 2 of the temperature fall on the shoulders and the difference in temperature between reference ceiling and cold ceiling, for all the tests concerned. Putting aside the tests marked x, where the shoulder temperature did not go down to the linearly falling part of the *cold* side of the critical temperature zone, 32 to 32.5 C, it is then found that a 1 C deg temperature increase on the ceiling means 0.20 to 0.24 deg fall of the shoulder temperature with naked upper body, whereas the corresponding value with a layer of clothes over the shoulders amounts to 0.16 to 0.24 deg and with 2 to 3 layers of clothes to 0.09 to 0.13 C deg.

INVESTIGATION IN AN UNDERGROUND WORKSHOP

Opportunity was provided to verify experimentally on a practical scale the relations found between air and ceiling temperature with ceiling cooling in an underground factory (Svenska Flygmotor, Trollhattan). The premises consist of an 11 m (36 ft) wide main shop with average ceiling height 6 m (20 ft). The space angle of the cold ceiling to a person standing in the middle of the shop would therefore be the same as prevailed for the laboratory tests carried out on a model scale. The ceiling temperature was taken thermoelectrically at 19 separate points, giving a mean value of 16.3 C (61.3 F), while the air temperature was kept to 18.3 C (64.9 F) at breast height above the floor. The subjective sensation of the degree of warmth in the space may best be characterized as *agreeably cool*, with a slight transient feeling of draft over the shoulder and head on coming into the premises. These conditions obviously lie wholly within the comfort zone according to the chart in Fig. 6. On that chart it can be read that these temperature conditions correspond to skin temperatures on the shoulders between 32.5 and 33 C. It may also be seen from the chart that a lowering of the ceiling temperature to about 14 C (57.2 F) requires to be compensated by a raising of the air temperature in the premises to 20 to 21 C to be able to retain the same shoulder temperature. It should, however, be stated that the workers in this workshop are seldom exposed to the cold radiation from the cold ceiling that corresponds to the conditions in the model tests. This is due partly to the fact that the workers are standing as a rule close to one of the walls, which means a smaller space angle to the ceiling, and partly owing to their often being subjected to radiant heat from the machines operating on the premises.

SUMMARY

The influence of the temperature of the ceiling surface (with angle of space at 2.2 steradians) on the skin temperature of the upper body naked and clothed has been studied for air temperatures between 16 (60.8 F) and 28 C (82 F) on subjects at rest and at work. On moving from beneath a warm ceiling to beneath a cold ceiling, a rapid fall of temperature was observed on the body surfaces exposed to the ceiling. If a slight chill reaction developed (falling finger and toe temperatures) there was occasionally observed an increase in temperature of the trunk's surface setting in later. This is assumed to be due to the beginning of displacement of the circulating blood volume from the extremities to the trunk. Good correlation was obtained between the equilibrium values for skin temperature of shoulders and the mean of air and ceiling temperatures. A fall of the tempera-

TABLE 2—QUOTIENT BETWEEN TEMPERATURE FALL ON SHOULDERS (Δt_{sh}) AND TEMPERATURE DIFFERENCE BETWEEN REFERENCE CEILING AND COLD CEILING ($\Delta t_{ceil.}$)

TEST No	AIR TEMP.	OTHER CONDITIONS	$\Delta t_{ceil.}$	Δt_{sh}	$\frac{\Delta t_{sh}}{\Delta t_{ceil.}}$
1	18.8	Upper body naked, at rest	5.5	1.1	0.20
2	19.3	Upper body naked, at rest	6.7	1.6	0.24
3	25.0	Upper body naked, at rest	11.2	2.3	0.21
5	26.0	Upper body naked, at rest	9.5	1.3	0.14[a]
6	28.5	Upper body naked, at rest	17.0	1.4	0.08[a]
7	16.6	1 cloth. layer, at rest	3.7	0.9	0.24
19	19.0	1 cloth. layer, at rest	5.1	0.8	0.16
8	21.0	1 cloth. layer, at rest	9.0	1.5	0.17
9	23.0	1 cloth. layer, at rest	8.5	1.4	0.16
11	26.2	1 cloth. layer, at rest	14.0	1.6	0.11[a]
12	25.0	2 cloth. layers, at rest	11.5	1.0	0.09[a]
13	15.6	2 to 3 cloth. layers, at work	4.6	0.4	0.09
14	17.5	2 to 3 cloth. layers, at work	5.2	0.7	0.13
15	18.0	2 to 3 cloth. layers, at work	6.0	0.4	0.07[a]
16	20.0	2 to 3 cloth. layers, at work	7.5	0.0	0.0[a]
17	22.2	2 to 3 cloth. layers, at work	10.6	0.4	0.04[a]
18	24.0	2 to 3 cloth, layers, at work	10.6	0.9	0.09[a]

[a]Shoulder temperature on the *warm* side of the critical temperature zone (32 — 32.5C).

ture of the ceiling surface by 1 C deg led to a fall of shoulder temperature of 0.20—0.24 C deg on naked upper body at rest; 0.16 to 0.24 C deg with one layer of clothes and rest; and 0.09 to 0.13 C deg with 2 to 3 layers of clothes and light work. On the basis of the results attained, a *comfort chart* has been drawn for the ceiling surface and air temperatures which may be expected to give the same shoulder temperatures.

REFERENCES

1. Temperature Changes in Blood Flowing in Arterials and Veins in Man, by H. C. A. Bazett (*Journal Applied Physiology*, Vol. 1, 1948, p. 3).

2. Basal Metabolism, Radiation, Convection and Vaporization at Temperatures of 22 to 35 C, by J. D. Hardy and E. F. Du Bois (*Journal Nutrition*, Vol. 15, 1938 p. 477).

3. Physiology of Heat Regulation and the Science of Clothing, by L. H. Newburgh, Editor (W. B. Saunders Co., Philadelphia, 1949).

4. Studies on the Heat Loss by Radiation and Convection from the Clothed Human Body, by M. Nielsen and L. Pedersen (*Acta Physiol. Scand.*, Vol. 27, 1952, p. 272).

5. Varmestralingsundersogelser, by L. Pedersen (Copenhagen, 1948 Diss.).

6. Varmeafgivelse fra en paklaedt termostat ved forskellige luft—og stralingstemperaturer, by L. Pedersen (*Teknologisk Institut*, Varmeteknisk afdeling, 1926-1951, Copenhagen, 1951, pp. 49-81).

7. Lamplig varmegrad och nodig luftvaxling, by H. E. Ronge (*Byggmastaren* B10, 1952, pp. 205-210).

8. A New Method of Partitional Calorimetry, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, Vol. 116, 1936, p. 641).

9. Temperature and Human Life, by C.-E. A. Winslow and L. P. Herrington (*Princeton University Press*, 1949).

1633

No. 1633

LOCAL CLIMATIC WEATHER DATA

By W. L. HOLLADAY,* LOS ANGELES, CALIF.

SOUTHERN California includes coastal, agricultural, metropolitan and desert localities. Design temperature data are published in The ASHAE GUIDE[1] (Reference 1, Chapter 12, Table 1, p. 251) for 5 cities in this area, *viz:* Bakersfield, Burbank, Daggett, Los Angeles and San Diego. The winter climate varies from Coronado near San Diego, where frost is unknown, to Bodie in the High Sierras where −36 F has been recorded. The summer climate ranges from an August normal daily maximum of 69 F in Santa Monica to 116 F for the equivalent figure at Greenland Ranch in Death Valley, whose record reading of 134 F stands as the official all-time high in the United States.

It is apparent that the heating and cooling loads must vary widely, sometimes from town to town. For example, the airline distance between Santa Monica and Los Angeles is 14 mi, and the summer temperatures change slightly more than 1 deg per mile in that distance.

All known applicable Weather Bureau publications were consulted (for list, see References and Bibliography). In addition, study was given to the recommended design temperatures published by this Society, the *ASRE,*[2] and several manufacturers, distributors and engineering offices. For summer data, the publication, *Summer Weather Data*[3] and the Air Force *Compilation of Air Weather Service Weather Data*[4] were drawn on heavily. Tabulations from log sheets of private observers, fire stations and packing houses were used.

Unpublished data from the files of the Fruit Frost Service[5] in Pomona were utilized in checking winter isotherms.

Data from so many sources naturally developed some inconsistencies, but surprisingly few cases of large differentials, say 5 deg or more. Much more typical was the case of Reno, Nevada, whose recommended summer design dry-bulb temperatures from several sources were 96, 94, 92, 93 and 95 F.

SCOPE OF WORK

It is the responsibility of the Society committees to develop standards of design, such as the TAC 2½ percent method of establishing a summer design dry-bulb

---

*Partner, Holladay & Westcott, Member of ASHAE.

[1]Exponent numerals refer to References.

Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, January 1958.

TABLE 1—WINTER DESIGN BASES FOR SOME WESTERN CITIES

(All figures in Fahrenheit degrees)

STATE	STATION	COMMON USE	TAC 97½%	THOM[6] 0.075
Ariz.	Phoenix	25	31	36
Calif.	Fresno	25	32	32
	San Diego	35	43	43
Colo.	Denver	−10	0	−12
	Pueblo	−20	2	−14
Idaho	Boise	−10	5	−10
	Pocatello	− 5	6	−17
Mont.	Billings	−25	−17	−31
	Miles City	−35	−18	−35
Nev.	Reno	− 5	0	3
N. Mex.	Albuquerque	0	16	8
Ore.	Portland	10	22	10
Utah	Salt Lake City	−10	7	− 1
Wash.	Seattle	15	24	15
	Spokane	−15	4	−16
Wyo.	Cheyenne	−15	− 3	−19
Averages		2.5	9.6	−0.5

temperature. The assignment of the Task Force was to find methods, preferably simple ones, of developing *equivalent* design temperatures: methods which, when applied to cities for which official temperatures had been developed, would give the same answers; and to do this for as many towns as possible. This scope ruled out time-consuming statistical methods; no computor was available. It was decided not to limit the study to larger towns, for a crossroads of today may be an aggressive building development tomorrow. It was decided also to include nearly every locality where a weather station had ever been located, even including a few spots no longer inhabitated, for they helped to develop data for nearby thriving communities without weather stations.

The original tables were developed from Weather Bureau figures up to 1930. Where later data were published in Annual Summaries, these were also taken into account, particularly for temperatures showing highest or lowest of record. It is interesting that for the towns having pre-1930 records, 51 new highs and 55 new lows have been recorded since 1930. (Summers are getting hotter and winters are getting colder.)

WINTER DESIGN DATA

In general, 3 bases for winter data were available: (1) Common Use, (2) the TAC 97½ percent basis, and (3) the Thom[6] 0.075 figures. A comparison of these for some western cities is shown in Table 1.

Although the choice of cities may seem arbitrary, it is simply that those listed are all the western cities for which all 3 bases were available. The average of the Thom[6] 0.075 scale is 3 deg below the average of Common Use, while the TAC 97½ percent scale is 10.1 deg above the Thom[6] 0.075 and 7.1 deg above Common Use. The Thom 0.075 scale was deliberately chosen by its author to approxi-

mate the level of Common Use, so the agreement is not surprising. The fact that the TAC 97½ scale gives such high readings is a reason for its unpopularity. It is reported that the scale is not being used because experienced engineers think it does not give enough factor of safety for equipment sizing. Hence, there is a need to develop a scale which gives the same general level of answer as either the Thom 0.075 or Common Use.

In 1947[7] the author offered the hypothesis that a winter design temperature could be developed by some relationship between the average daily minimum temperature in the coldest month (January, without exception) and the lowest temperature of record. Since both these figures were published for all Weather Stations, efforts were made to correlate them both to the Thom 0.075 base and the Common Use. The correlations were not satisfactory for some or all of the following reasons:

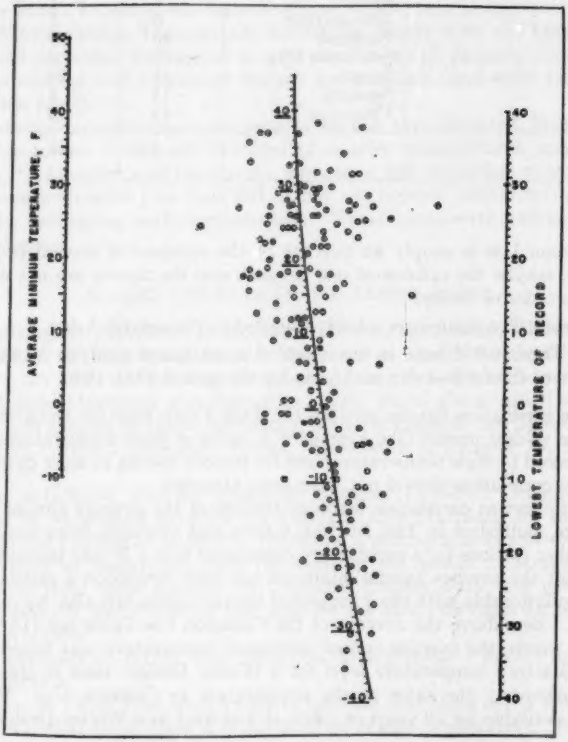


FIG. 1—WINTER DESIGN TEMPERATURE NOMOGRAPH. POINTS REPRESENT AVERAGE ANNUAL MINIMUM TEMPERATURES

TABLE 2—COMPARISON OF TASK FORCE RECOMMENDATIONS WITH OTHER BASES
(All figures in Fahrenheit degrees)

STATE	STATION	COMMON USE	AVGE ANNUAL MIN	TASK FORCE RECOMMENDED DESIGN
Ariz.	Phoenix	25	26	25
Calif.	Fresno	25	26	27
	San Diego	35	37	38
Colo.	Denver	−10	−11	−12
	Pueblo	−20	−13	−11
Idaho	Boise	−10	− 1	−10
	Pocatello	− 5	−12	− 8
Mont.	Billings	−25	−30	−28
	Miles City	−35	−30	−30
Nev.	Reno	− 5	− 1	− 2
N. Mex.	Albuquerque	0	2	3
Ore.	Portland	10	18	14
Utah	Salt Lake City	−10	− 2	− 3
Wash.	Seattle	15	20	18
	Spokane	−15	− 5	−11
Wyo.	Cheyenne	−15	−18	−19
Average....		−2.5	+0.4	−0.6

(1) Common Use is simply an average of the opinions of several people (in a small town, maybe the opinion of one person), and the figures are not necessarily of the same order of verity.

(2) Common Use figures are usually rounded to the nearest 5 deg.

(3) The Thom 0.075 base is the result of a statistical analysis of the average temperature of the coldest *day* each year for the period 1921–1950.

Since the correlation figures used by the Task Force were for the average minimum of the coldest month (the average of a series of *night* temperatures) and the lowest of record (a *night* temperature) and for periods ending in most cases in 1930, satisfactory correlation should not have been expected.

The next effort at correlation involved the use of the average annual minimum temperature, published in THE ASHAE GUIDE and available from enough of the larger weather stations for a satisfactory correlation test. It was learned from the attempt that the average annual minimum not only developed a satisfactory and consistent relationship with the 2 suggested temperatures, but also, by coincidence, averaged 1.1 deg above the average of the Common Use figure for 115 test cities.

In other words, the average annual minimum temperature was found to represent a satisfactory temperature level for a Winter Design, since in the aggregate it is approximately the same as the temperature in *Common Use*. Where this figure was available for 10 years or more, it was used as a Winter Design Temperature.

Several methods of weighting the January average minimum temperature and the lowest of record were tried, but it was found that the weighting factor required to obtain the Average Annual Minimum was not constant, but varied with the

design temperature level: the colder the climate, the more important the lowest of record became. To avoid burdensome arithmetic, a graphical solution was employed, as shown in Fig. 1. To use Fig. 1 to determine a Winter Design, a straight edge is placed over the correct figures for Average Minimum Temperature and Lowest Temperature of Record, and one reads the recommended design temperature on the center curve.

Due to the nomograph construction, the deviation are smaller than the point clusters would indicate. Temperatures are read on a vertical scale, and the deviation of any point from the curve is approximately the tangent of the angle the straight edge makes with the horizontal, which runs from about 16 to 33 deg. Hence the average deviation is from 0.29 to 0.64 of the horizontal distance of point from the line. The worst case investigated is that of Miami, Fla., whose average annual minimum is 38 F; but the recommended design temperature by this method would be 45 F. The correlation of this method for the western cities tabulated in Table 1 is given as Table 2.

In other words, a method has been developed which, very simply, yields a recommended Winter Design Temperature correlating closely with the accepted level. The method may then be applied to any city for which January Average Minimum Temperature and Lowest of Record are available, and some faith may be placed in the result.

For those communities where no weather station ever operated, Winter Design Temperatures were chosen by averages of nearby temperatures, sometimes by isotherms. A knowledge of the terrain, elevations and prevailing wind, as well as acquaintance with local jobs that did or did not operate satisfactorily, were invaluable in estimating such temperatures. These cases were estimated to the nearest 5 degrees.

SUMMER DRY-BULB DESIGN TEMPERATURES

With satisfactory experience in the use of the Average Annual Minimum, it was natural to turn to the Average Annual Maximum as a guide for the average level of summer dry-bulb design temperatures. A very few trials showed that the Average Annual Maximum was from 5 to 10 deg above the accepted design dry-bulb level, and showed no useful correlation with, for example, the *Summer Weather Data*[3] or THE GUIDE 2½ percent figures. (Reference 1, Chapter 13, Table 2, p. 279).

Again, it is believed that an acceptable Summer Dry-Bulb Design Temperature should lie somewhere between the Average Daily Maximum for the warmest month† and the Highest of Record. Therefore, a nomograph was plotted (Fig. 2) with these lines as boundaries, lines representing the spread of accepted Design Temperatures from these sources, *viz: Summer Weather Data*[3] 2½ percent, THE GUIDE 2½ percent, and Common Use, and for the about 60 cities for which these 3 data are available.

Several examples of rather wide variation among these 3 figures appeared as shown in Table 3.

The figures in Table 3 are shown only to indicate that there exists a diversity of opinion as to what constitutes a good summer design temperature. Where two different 2½ percent figures occur, it must be because the source data came from

†Unlike the Average Daily Minimum, which always occurs in January, the Average Daily Maximum in the West may occur in any month from June through September.

different locations or from different years. The Guide 2½ percent data are from airports; in some cases this may make a considerable difference.

The average of the 67 *Summer Weather Data* 2½ percent numbers from Reference 3 is 93.9 F. The average of the sixty 2½ percent numbers in The Guide is 93.2, and the average of the Common Use number is 93.7. Therefore, while local differences of opinion or data certainly exist, there is good agreement as to what the general level should be. If one takes figures from the dotted vertical line on the nomograph, Fig. 2, the average of 60 cities is 93.1, and the resulting design temperature lies on the line (that is, within the range of the 2 or 3 presently accepted numbers) for 39 out of the 60 cities. For another 8 cities, the nomograph gives an answer within 1 deg of one of the accepted figures. For the other 13 cities, the deviation is from 2 to 5 deg, either up or down. Two to 5 deg is almost typical of the spread among the three *accepted* figures.

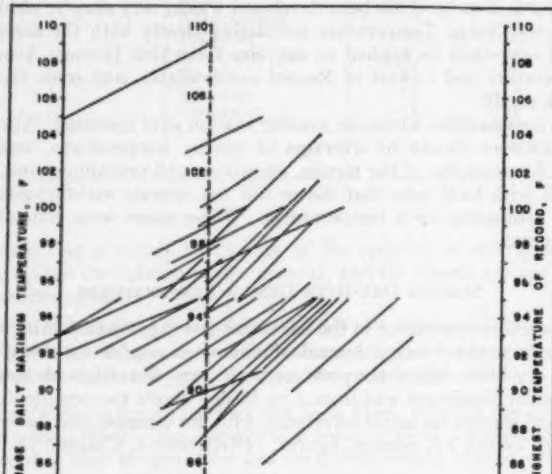


Fig. 2—Summer Design Dry-Bulb Nomograph. Lines Represent Range of Suggested Design Dry-Bulb Temperatures for Given Cities

TABLE 3—UNUSUALLY WIDE VARIATION AMONG ACCEPTED SUMMER DRY-BULB DESIGN TEMPERATURES

(All figures in Fahrenheit degrees)

	COMMON USE	*Summer Weather Data* 2½%	GUIDE 2½%
San Diego	85	81	79
Detroit	95	93–	89
Helena	95	89	
Newark	95	91	89
Buffalo	93	87	86
Pittsburgh	95	91	88
Brownsville	100	92	

The dotted vertical line on the nomograph is an effort to correlate the somewhat conflicting data in terms of the parameters chosen, namely: Average Daily Maximum Temperature and Highest Temperature of Record. A simple weighted average can be used instead of the nomograph, thus:

Design Dry-Bulb Temperature = [2 (Av Daily Max) + (Highest of Record)]/3

SUMMER WET-BULB DESIGN TEMPERATURES

Finding wet-bulb design figures proved to be the most difficult of the problems faced. It was approached from several directions.

Many weather stations publish relative humidities at or near noon. Simultaneous dry-bulb temperatures are also available, from which average noon wet-bulb temperatures were obtained. If a horizontal line is drawn on a psychrometric chart from this average noon dry-bulb and wet-bulb temperature to the summer Dry-Bulb Design Temperature, a *Wet-Bulb Design Temperature* may result* (see Fig. 3).

Summer Weather Data[3] includes a table, *Analysis of Wet-Bulb Temperatures on a 24-Hr Basis*, listing hours *vs* wet-bulb temperature for the months June through September for 67 U. S. cities. If these data are plotted on 3-cycle semi-log paper, slopes are developed which are characteristic of the general locality (mountain, coastal, etc.). The TAC recommends that the wet-bulb design temperature be taken as that which is exceeded during 5 percent of the hours for the 4 summer months: actually, 146.4 hr for the *Summer Weather Data*.

The Air Weather Service has published for many airport localities, both domestic and foreign, the number of hours for 6 summer months when wet-bulb temperatures exceed the 2 levels of 67 F and 73 F. When these points are plotted on the same chart as the *Summer Weather Data*,[3] and the latter characteristic curve shape is matched, interpolation to 146 hr is possible. In some cases, judicious extrapolation may be used. The *Summer Weather Data* period is 2928 hr and the Air Weather Service period is probably 5160 hr—the 6 *warmest months* may not be the same for each location. It is believed that the change of hours for given wet-bulb levels by the addition of 2 fall or spring months is likely to be slight, and

*The method was devised by Douglas Hotes, a member of the Task Force.

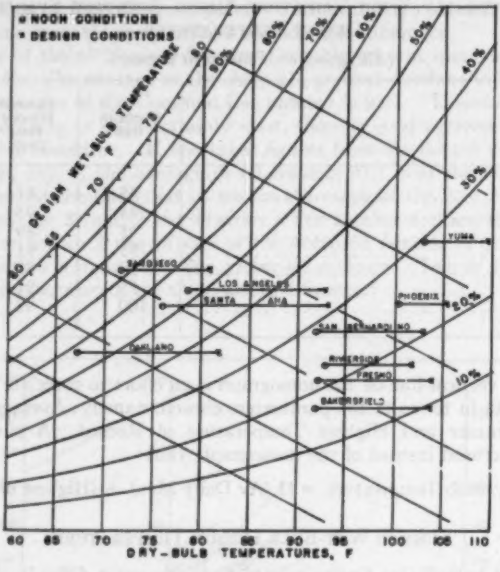


FIG. 3—ONE METHOD OF OBTAINING DESIGN WET-BULB TEMPERATURES

some random tests for cities having both *Summer Weather Data* and Air Weather Service data show good correlation. Sample plots are shown in Fig. 4.

Again, the psychrometric chart method was tested by comparison with the *Summer Weather Data* 5 percent figures, with the Air Weather Service 146-hr interpolations, and with the Design Wet-Bulb Temperature in Common Use from THE GUIDE. The comparison is shown in Table 4.

The final method used was to spot Wet-Bulb Design Temperatures believed to be trustworthy on a large scale map and to interpolate intermediate points, correcting for elevation, presence of bodies of water, and terrain where necessary.

Arguments on wet-bulb design temperatures can generate more heat than any kindred subject. The numbers here presented are the best the Task Force was able to hammer out, and that's all that is claimed for them. There is, however, one discussion that maybe can be resolved: whether or not the wet-bulb temperatures are increasing due to greater usage of water. The official position of the Weather Bureau, as taken from Local Climatological Data, Phoenix, Ariz., 1956 (and previous years) is:

The central floor of the Salt River Valley is irrigated by water from dams built on the Salt River system. To the north and west of the gravity flow irrigated district there is considerable agricultural land irrigated by pump water. There is no evidence that the irrigation has in any way affected the relative humidity in the valley. The average daytime relative humidity is 29 percent based on observations at 11:30 a.m. and 5:30 p.m.

This present investigation does not reach the same conclusion. A comparison of figures for number of hours at and above given wet-bulb temperatures (*Summer Weather Data*[1] Method) for Phoenix shows the figures in Table 5.

Therefore, if these years, shown in Table 5, are typical, the design wet-bulb temperature in Phoenix is increasing at the rate of 0.03 F deg per year.

GENERAL

Preliminary copies of the findings of the Task Force were presented to the Chapter's Research Committee on June 4, 1957. At this writing not enough time has elapsed to determine whether the volume of disagreements will be small or large. Surely some changes and needs for improvement will develop with the use of the tables. The Task Force has been reappointed as a standing committee to review requests for changes and additions and to incorporate them into the next printing of the report. Table 6 is a sample page of the completed 16-page report.

Where the need for local climatic weather data exists, as must be the case in many coastal and mountain areas, this kind of project can be instituted by local chapters as a part of the continuing effort of the profession to improve design information.

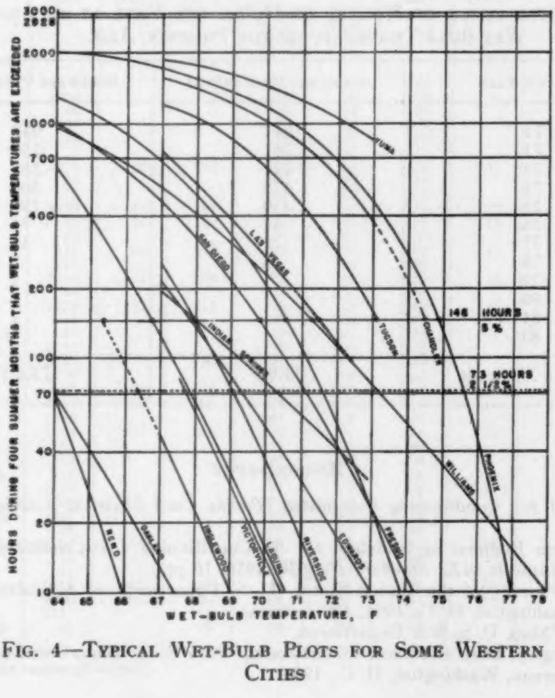


FIG. 4—TYPICAL WET-BULB PLOTS FOR SOME WESTERN CITIES

TABLE 4—TYPICAL WET-BULB DESIGN TEMPERATURES

STATE	STATION	COMMON USE	*Summer Weather Data*[3]	AIR WEATHER SERVICE	SO. CALIF. TASK FORCE
Ariz.	Chandler		74[a]	76	75
	Flagstaff	65	67[a]		66
	Phoenix	76	75	75	76[b]
	Tucson	72	72[a]	73–74	73
	Yuma	78	80[a]		78
Calif.	Bakersfield	70	68[a]		70[b]
	Fresno	74	70	70	70
	Inglewood		67[a]	66	67
	Los Angeles	70	67[a]		71
	March Field		69[a]	70	68
	Oakland	65	63	Below 64	63
	Oxnard		65[a]	66	66
	Riverside		68[a]	68	71
	San Bernardino			72	72
	San Diego	68	70	68	71
	Santa Ana			69	70
Nev.	Las Vegas	75	72[a]	72–73	72
	Reno	65	62	63	62
	Winnemucca	65	61[a]		61

[a]Indicate interpolation from contours, *Summer Weather Data* Chap. 7, Fig. 1.
[b]These figures obtained by statistical analysis of recent years.

TABLE 5—COMPARISON OF NUMBER OF HOURS PER YEAR AT OR ABOVE GIVEN WET-BULB TEMPERATURES FOR PHOENIX, ARIZ.

WET-BULB, FAHR.	HOURS PER YEAR 1930-34	HOURS PER YEAR 1953-56
71	941	942
72	741	758
73	472	552
74	275	366
75	130	195
76	44	74
77	10	18
78	3	3
79		1
80		1
81		1
82		1
Design 5% basis	74.8 F	75.4 F

BIBLIOGRAPHY

Report of Air Conditioning Committee (*Pacific Coast Electrical Association*, 1939–40, 5 pp.).

Application Engineering Standard for Air-Conditioning (*Air-Conditioning and Refrigeration Institute, ARI Standard No.* 530, 1956, 16 pp.).

Climatic Survey of the United States (U. S. Department of Agriculture Weather Bureau, Washington, D. C., 1934, 106 sections).

Isotherm Map, U. S. War Department.

Climatological Data and Local Climatological Data (U. S. Department of Commerce Weather Bureau, Washington, D. C., 1956).

Design Weather Data (Fairbanks, Morse & Co., Chicago, 1946, 6 pp.).
A Study of Air Conditioning Outdoor Weather Design Data for Selected California Cities, by John H. Grim (Thesis for degree of B.S., California State Polytechnic College, San Luis Obispo, 1956, 55 pp.).
Outdoor Design Temperatures—California, Arizona and Nevada Sections (Holladay & Westcott, Los Angeles, 1955, 8 pp.).
Tentative Design Temperatures (Reliable Steel Supply Co., Los Angeles, 1956, 9 pp.).

REFERENCES

1. HEATING VENTILATING AIR-CONDITIONING GUIDE 1956 (published by AMERICAN SOCIETY OF HEATING AND AIR CONDITIONING ENGINEERS, New York, N. Y.).
2. *Air Conditioning Refrigerating Data Book* (published by *The American Society of Refrigerating Engineers*, New York, N. Y.).

TABLE 6—SAMPLE PAGE FROM SOUTHERN CALIFORNIA DESIGN TEMPERATURE REPORT

	Tentative Outdoor Design Temperatures								
		Summer					Winter		
	Elevation, ft	Avg Daily Max	Highest of Record	Rec. Design		Outdoor Daily Range	Avg Daily Min	Lowest of Record	Recommended Design
				Dry-Bulb	Wet-Bulb				
La Puente	320			*100*	*70*	*32*			*30*
La Verne	1050			*100*	*69*	*36*			*30*
Lawndale	66			*85*	*67*	*16*			*35*
Lemon Grove	437			*95*	*72*	*22*			*30*
Lennox	71			*85*	*67*	*16*			*35*
Lido Island	7			*80*	*68*	*12*			*35*
Llano	3820	95	112	101	72	33	32	8	19
Lomita	56			*90*	*67*	*14*			*35*
Lompoc	72			*90*	65	*20*			30
Lone Pine	3950	94	105	98	65	39	24	2	11
Long Beach	34	80	108	89	68	19	42	21	33
Los Alamitos	20			*90*	*69*	*20*			*35*
Los Angeles	312	84	110	93	71	22	45	28	37
Los Nietos	194			*95*	*70*	*24*			*35*
Lundy Lake	7760	*77*	90	81	63	27	16	−34	−16
Lynwood	88			*90*	*70*	*20*			*35*
Lytle Creek	2250	91	111	98	69	39	44	19	31
Malibu	18			*80*	*64*	*14*			*35*
Manhattan Bch.	120			*80*	*66*	*14*			*35*
March Field	1534			98	68	*38*			*25*
Maricopa	680		116	*105*	*67*	*34*		19	*26*
Maywood	170			*90*	*70*	*22*			*35*
Mecca	−185	106	126	113	76	30	37	13	24
Mill Creek No. 2	2965	91	108	97	70	26	41	16	28
Modjeska	1240			*95*	*70*	*32*			*25*
Mojave	2775	*103*	118	108	71	34	*30*	10	19
Monrovia	562			*95*	*70*	*30*			*30*
Montebello	205			*95*	*70*	*24*			*35*
Monterey Park	380			*95*	*71*	*26*			*30*
Montrose	1224			*95*	*71*	*34*			*30*
Moorpark	500			*100*	*72*	*38*			*25*
Morongo Valley	2587			*105*	*73*	*34*			*25*
Moneta	36			90	67	20			35
Mount Wilson	5709	84	100	89	67	21	34	7	17
Naples	26			*85*	*68*	*18*			*35*
National City	34			*90*	*71*	*12*			*35*
Needles	913	108	125	114	75	28	39	18	27
Nellie	5000	84	100	89	67	27	31	9	19
Newhall	1243		113	*100*	*71*	*38*		10	*20*
Newport Beach	10		99	*80*	*68*	*14*		31	34

Italics indicate estimated values.

3. *Summer Weather Data* (The Marley Co., Kansas City, 1939).
4. Air Weather Service Weather Data (Directorate of Facilities Support, Office of the Assistant Chief of Staff, Installations, Hq USAF, Washington, D. C., 1954, 39 pp.).
5. Various unpublished data, U. S. Department of Agriculture, Fruit Frost Service, Pomona.
6. Revised Winter Outside Design Temperatures, by H. C. S. Thom (ASHAE TRANSACTIONS, Vol. 63, 1957, pp. 111–28).
7. Determination of Outside Design Temperatures — A Suggested Approach, by W. L. Holladay (ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, July 1947, pp. 117–22).

DISCUSSION

R. G. VANDERWEIL, Boston, Mass., (WRITTEN): Being accustomed to design problems of the eastern U. S. only, I would prefer not to comment on the *numerical* results of the method proposed, but rather to remark in general. As consultants, no doubt, we are in need of reliable and rational design data based on up-to-date statistical methods rather than on *an average of opinions* obtained some time ago. The attempts of Mr. Holladay first to carefully tabulate and graphically list statistical data, and then to average and interpret this data consistent with the findings, and still do so in such a manner as to result in recommendations close to those formerly used as *average opinions*, is no doubt an attempt as good as any to rationalize our design basis.

No matter how we may attempt to find solutions to the problem of design temperatures, we will always be confronted with the question of how close a temperature should we select to the average and how close to the extreme (winter and summer) conditions. In view of the fact that previous GUIDE design data (in the East) has given consistently satisfactory results, I fully agree with the author that any method based on statistical data should not deviate too far from formerly practiced values. That is, at least for heating.

In regard to cooling, I have noted that heat gains calculated by sales offices of equipment manufacturers are fairly consistently lower than those calculated by engineering offices. Whether this might suggest slight upward revision of internal or downward revision of external design temperature, or simply should be considered indication of keen competitive bidding, I should not like to judge at this time. However, it is my feeling that we are quite safe in our heat gain calculations.

W. A. DANIELSON, Memphis, Tenn.: I wonder if Mr. Hess could tell us how much effect the smog had to do with this?

A. J. HESS, Los Angeles, Calif.: It didn't have much to do with the design temperature, but it sure does have a lot to do with the design conditions.

AUTHOR'S CLOSURE: Thanks are extended to Mr. Vanderweil for his kind comments, with which the author is in substantial agreement.

General Danielson's question was well answered by Mr. Hess, but possibly the comment may be added that since the Los Angeles smog was born in Detroit, the problems raised by it may well have to be solved there.

1634

No. 1634

A DYNAMIC HEAT STORAGE SYSTEM

By T. L. ETHERINGTON*, SCHENECTADY, N. Y.

WIDESPREAD use of air-source heat pumps is dependent upon favorable electric power rates as determined by the cost to serve this load. Large amounts of electrical energy cannot be stored like combustible fuels and, therefore, a large intermittently operated load such as the heat pump becomes, under certain climatic and application conditions, an expensive load to serve.

The capacity (and electrical demand) of the air-source heat pump is greatest when its heat requirement is least and vice versa. Therefore, it is conceivable that its excess heat could be stored during low heat demand periods for use during high demand periods. The result would be a smaller unit (reduced electrical load) approaching continuous operation. This would constitute a more desirable load to serve, and should result in reduced rates for electrical service.

The general requirements for a heat storage system are low cost, reliability, safety and compactness. The compactness requirement indicates that storage as latent rather than sensible heat is preferable.

A disodium phosphate-water system has undergone considerable evaluation as an attractive possibility for heat storage due to its cheapness and the relatively high heat of fusion of the dodecahydrate ($Na_2HPO_4 12H_2O$). Its melting point of 96 F would also permit heat pumps to store their excess capacity during periods of low demand.

Unfortunately disodium phosphate does not always crystallize as the dodecahydrate (or 12-hydrate), but may form a heptahydrate which has a much lower heat of fusion. Previous investigators have attempted to eliminate formation of the $Na_2HPO_4.7H_2O$ (heptahydrate or 7-hydrate) phase by high temperature treatment to destroy seed crystals of the 7-hydrate, followed by rapid quenching to make the system metastable with respect to the heptahydrate[1]. Subsequent melting and freezing cycles have always been found to result finally in the formation of some of the 7-hydrate and reduced effectiveness of the storage salt.

The usual method of application has been to enclose the heat storage salt in a sealed container around which a heat exchange fluid is circulated[2]. This tech-

---

* Chemical Process Research, Research Laboratory, General Electric Co.

[1] Exponent numerals refer to References.

Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, January 1958.

nique has several disadvantages. For example, during freezing (release of stored energy) the heat transfer rate falls off rapidly as solids build up on the container walls, and eventually becomes too slow for practical applications. However, when this point is reached, there may well remain a core of liquid within the container, representing a large fraction of the stored heat. A second disadvantage is the undesirable subcooling without crystallization which can occur if the storage salt containers are cooled without provisions to insure nucleation.

This paper describes a new technique for utilizing the heat of fusion of a storage material. The system is termed dynamic, since the heat transfer fluid flows through

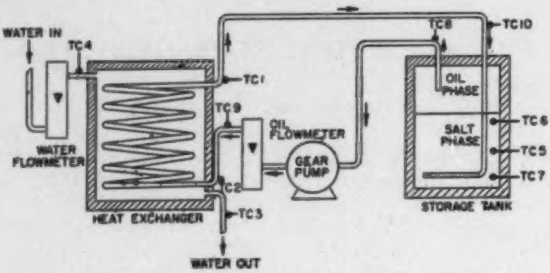


FIG. 1—DYNAMIC HEAT STORAGE APPARATUS

the salt to provide agitation plus direct contact heat exchange. Disodium phosphate dodecahydrate ($Na_2HPO_4.12H_2O$) was used for the storage material.

OBJECTIVES

The objectives of these experiments were as follows:

1. To demonstrate the feasibility of the dynamic heat storage system;
2. To assess the effect of agitation upon subcooling of the salt melt;
3. To assess the effect of direct-contact heat exchange upon heat transfer rates;
4. To determine whether agitation of the disodium phosphate dodecahydrate would prevent crystallization of undesirable heptahydrate.

EQUIPMENT AND PROCEDURE

The heat storage system consisted basically of 2 heat exchangers and a circulating pump. A light mineral oil was circulated through a *usage* heat exchanger where heat was transferred from tap water to oil and from oil to tap water. The oil then bubbled through a *storage* heat exchanger in which it absorbed or rejected heat from or to the heat storage salt. Fig. 1 is a sketch of the apparatus.

Two layers existed in the storage tank with the oil as the top layer and the storage salt as the bottom layer. Oil was pumped from the top layer, through the heat exchanger (where it was heated or cooled by domestic hot or cold water) and discharged through a perforated pipe at the bottom of the salt layer. The oil absorbed or rejected heat as it rose through the storage salt phase.

Each run consisted of a freezing cycle starting from a melt temperature of about 130 F, immediately followed by a melting cycle to restore the salt to 130 F. The freezing cycle was considered complete when the temperature of the oil leaving the salt container reached about 80 F. The time between runs usually varied from 18 to 72 hr with the system being held at 130 F between each run. Runs 63 and

TABLE 1—PERCENT UTILIZATION OF THE HEAT OF FUSION OF $Na_2HPO_4.12H_2O$

RUN NO.	CYCLE (F = FREEZING; M = MELTING)	DURATION, MIN	HEAT REALIZED FROM SALT		LATENT HEAT UTILIZATION, PERCENT
			LATENT + SENSIBLE, BTU	LATENT, BTU	
59	F	130	5112	3791	80
	M	80	4361	3144	66
60	F	135	5056	3747	79
	M	95	3687	2259	48
61	F	230	5616	4311	91
	M	115	4929	3754	79
62	F	140	6395	5238	110
	M	105	5410	4174	88
63	F	155	5219	4108	87
	M	80	4015	2747	58
64	F	128	4326	3002	63
	M	80	3802	2709	57
65	F	145	4844	3587	76
	M	80	3373	2195	46
66	F	130	4531	3244	68
	M	90	4059	2757	58
67	F	185	5918	4590	97
	M	190	4933	3631	77
68	F	170	4723	3421	72
	M	155	5506	4144	87
69	F	185	4160	3066	65
	M	111	3705	2664	56
70	F	145	4152	2903	61
	M	115	4250	2948	62
71	F	170	4613	3266	69
	M	122	4525	3178	67
72	F	155	4000	2653	56
	M	85	4266	2893	61
73	F	195	4776	3478	73
	M	145	4579	3281	69
74	F	120	3399	2086	44
	M	100	2984	1750	37
75	F	140	3863	2897	61
	M	100	3816	2850	60
76	F	180	4241	2932	62
	M	125	3780	2418	51
77	F	135	3543	2404	51
	M	120	3115	2055	43
78	F	115	3634	2724	57
	M	130	3297	2308	49
79	F	155	4548	3216	68
	M	125	4313	3010	63
80	F	167	3924	2767	58
	M	115	3907	2750	58

64 made up 4 continuous cycles each immediately following the other. Runs 65–66 and 67–80 were also consecutive cycles with the storage salt being held at 130 F for 90 hr between run 66 and run 67 (Table 1).

Data were taken at 2 to 5 min intervals during the first 20 min of a cycle and 10 to 15 min apart thereafter. Thermocouple locations are shown in Fig. 1. A slot cut in the insulation of the glass storage tank permitted visual inspection of both salt and oil phases.

The storage salt was prepared by adding water to technical grade anhydrous sodium phosphate to make 40.2 lb salt of composition corresponding to the dodeca-

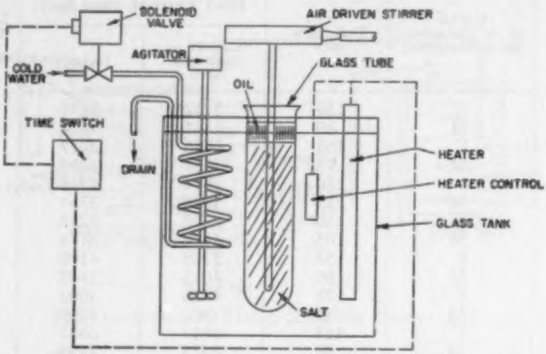


FIG. 2—APPARATUS FOR DETERMINING COMPLETENESS OF MELTING AND FREEZING OF $Na_2HPO_4.12H_2O$

hydrate. The mixture was heated at about 150 F to produce a clear melt before charging to the system. All runs were made using the same salt batch.

A separate experiment was made to assess the completeness of freezing and melting under conditions of more vigorous agitation than that supplied by the circulating oil. Salt of composition corresponding to the 12-hydrate was placed in a large glass test tube. A layer of light oil was added to the tube to provide a seal; an air-driven stirrer with a glass agitator was inserted in the tube, and the tube immersed in a water bath. A time switch actuated a cold water valve and resistance heater to provide a 3-hr cooling period followed by a 3-hr heating period. (See Fig. 2.) Cooling water temperature for freezing was 60 to 65 F. The bath temperature for melting was 130 F. The salt melted and froze completely for 220 six-hr cycles. The bath temperature for melting was lowered to 120 F and the salt again froze and melted completely for 40 six-hr cycles. The time required for complete melting at both bath temperatures was 45 min to 1 hr, about 20 min of which was required to raise the bath temperature from 60–65 F to the controlled bath temperature. Freezing required a total of 20–30 min, including the time required to lower the temperature from 120 or 130F.

CALCULATIONS

A heat balance was made for each cycle in the circulating system. Critical data for system components are listed in the Appendix. Symbols used in the calcula-

tions are as listed. Heat released from or stored in the salt was determined by graphical integration using oil flow rates and measured temperature difference between the oil entering and leaving the salt container.

$$H_1 = \Sigma H_0 - H_0 - H'_T \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

A second heat balance was made on the water to check the heat in the salt value obtained by the heat balance on the oil.

$$H_2 = \Sigma H_W \pm \text{Losses} \pm H_p - H_0 - H'_T - H'_x \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Heat input by the pump was determined similarly by using temperature differences in the oil between its exit from the storage tank and entrance to the heat exchanger. This value actually represents the net effect of pump inputs plus or minus gains or losses from ambient air temperature.

TABLE 2—SUMMARY OF CONTINUOUS RUNS 67–80[a]

CYCLE TYPE	DURATION (MIN)	HEAT TRANSFERRED TO WATER (BTU)	PUMP INPUT (BTU)	LOSSES (BTU)	LATENT HEAT STORED OR REMOVED (BTU)
Freezing	158	5462	528	30	2824
Melting	124	4946	40	80	2755

[a] Values shown are average.

Sensible heat quantities for system components were calculated using initial and terminal temperatures for each cycle. The sensible heat in the liquid salt was calculated by assuming that it all crystallized at 96 F. The lowest of the 3 measured final salt temperatures was used to calculate the sensible heat in the solid. The latent heat effect was then obtained by subtracting the sensible heat in the salt from H_1. Salt utilization percentages (Table 1) represent only that amount of latent heat realized based on a value of 118 Btu per lb for the heat[3] of fusion of $Na_2HPO_4.12H_2O$.

Runs 67–80, consisting of 14 complete and consecutive cycle pairs, allowed a comparison between successive freezing and melting cycles to determine whether the heat stored during the melting cycle was recovered during the following freezing cycle.

$$\Sigma H_W \text{ (freezing)} - \Sigma H_W \text{ (melting)} = H_0 \text{ (melting)} + H_p \text{ (freezing)} - \text{losses} \quad . \quad . \quad (3)$$

This comparison showed poor agreement between some cycle pairs, but when applied to the average values for 14 melting and freezing cycles the agreement was excellent. (See Table 2.)

DISCUSSION

The phase diagram for the disodium phosphate-water system is reproduced in Fig. 3. Two crystallization mechanisms are proposed. Consider first the case when heat is removed under equilibrium conditions from a melt of $Na_2HPO_4.12H_2O$. The solution cools to about 112 F at which point $Na_2HPO_4.7H_2O$ starts to crys-

tallize and liberate its heat of fusion. The solution becomes more dilute with temperature and composition following line A-B as the 7-hydrate crystallizes. As the temperature falls below point B, the 12-hydrate starts to crystallize and the 7-hydrate starts to pick up water to form also the 12-hydrate. (The latter process is much slower than the former.) Temperature continues to fall until all of the salt has crystallized. The reverse process takes place during melting.

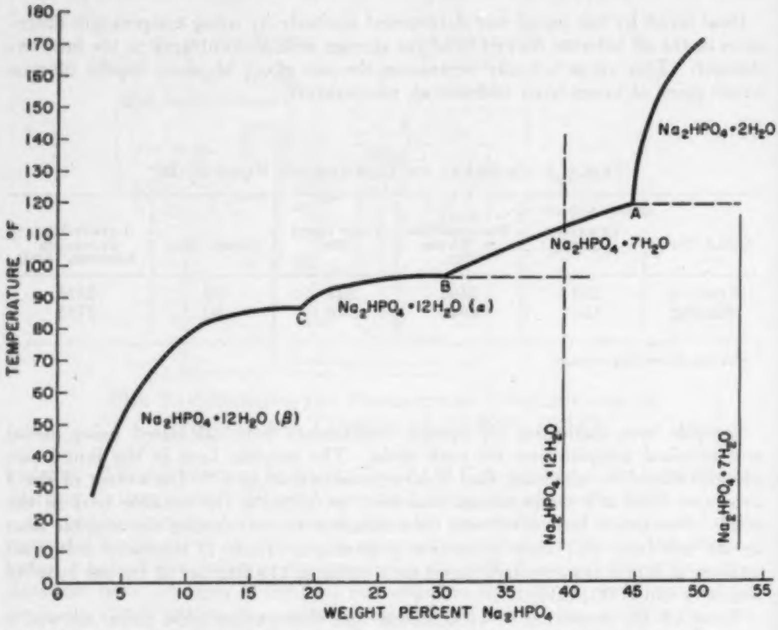


Fig. 3—Phase Diagram for the System Disodium Phosphate—Water

If the freezing process proceeds in this way, 56 percent of the salt crystallizes from the melt as the 7-hydrate, and 44 percent crystallizes as the 12-hydrate. The heats of fusion are 58 Btu per lb for the 7-hydrate and 118 Btu per lb for the 12-hydrate.[3] Therefore, it would be expected that the effective heat of fusion realized would be somewhere between 84 and 118 Btu per lb when the salt had completely crystallized from a melt. Assuming no 7- to 12-hydrate conversion, only 84 Btu per lb would be realized and about 0.14 lb of free water would remain when all the phosphate had crystallized. This would correspond to a utilization value of 71 percent.

A second mechanism of crystallization is possible when the system is metastable with respect to the heptahydrate. In this case the melt temperature can be success-

fully lowered through the region of heptahydrate stability without crystallization. Thus, the salt crystallizes entirely as the dodecahydrate. This would result in a salt utilization of 100 percent.

Agitation of the melt-solid mixture should increase the rate of conversion into 12-hydrate (Case 1) by avoiding large crystal agglomerates during that period when the 12-hydrate is crystallizing. More solid surface area would then be exposed for reaction of the heptahydrate with the water before the formation of large masses of blocking crystals.

The test tube experiment using a stirrer to provide violent agitation resulted in complete melting or freezing of the solid in less than 1 hr. The fact that the salt

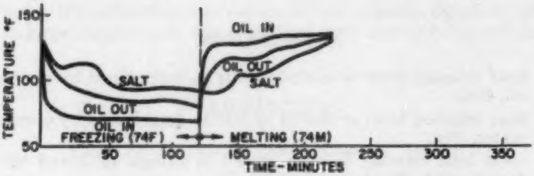


FIG. 4—TEMPERATURES FOR TYPICAL CYCLE RESULTING IN LOW VALUE FOR PERCENT SALT UTILIZATION

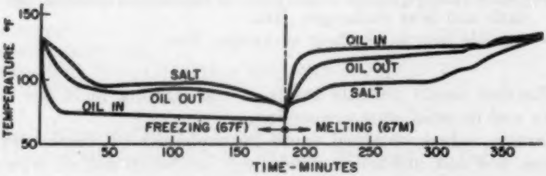


FIG. 5—TEMPERATURES FOR TYPICAL CYCLE RESULTING IN HIGH VALUE FOR PERCENT SALT UTILIZATION

froze completely at 96 F without residual water indicates that the ultimate solid was indeed the 12-hydrate (Case 2).

Observations and measurements for runs where agitation was produced by circulating oil were as follows.

1. Freezing run yields were greater following an overnight or longer heating at a melt temperature of 130 F.

2. Freezing runs in which salt utilization approached 100 percent indicated no crystallization occurring above 96 F. (See Figs. 4 and 5.)

3. At the point in a melting run when the melt temperature reached 130 F, small amounts of fine solids remained suspended in the melt. These solids were reduced in quantity but not always completely removed by overnight heating at 130 F.

Correlation of salt utilization percentages with crystallization mechanism for this system is questionable. Others[5] have reported the very slow solution of the 7-hydrate; further, if a melt containing even small amounts of heptahydrate crystals

is cooled, these crystals will seed the crystallization of more heptahydrate. Therefore, it is conceivable that the solids observed in the melt at 130 F were heptahydrate, which dissolved only after long heating at this temperature, and that more rapid solution of these crystals in the test tube was due to better agitation.

Since the latent heat utilization percentage obtained on freezing is a function of the crystallization mechanism and melt history, it reflects any erratic behavior of the storage material. If one assumes a clear melt which is metastable with respect to the heptahydrate, then a salt utilization percentage approaching 100 percent should be realized. Again, assuming a clear melt without metastability and without solid state conversion from the 7- to the 12-hydrate, then the utilization should be about 70 percent. Values between 70 and 100 percent might be due to lack of metastability but with some 7- to 12-hydrate conversion. Less than 70

NOMENCLATURE

H_1 = heat released from or stored in salt as determined by energy balance on oil, Btu.
H_2 = heat released from or stored in salt as determined by energy balance on water, Btu.
ΣH_0 = total heat released from or stored in storage tank and contents as determined by heat balance on oil, Btu.
ΣH_W = total heat transferred to or from water, Btu.
H_0 = sensible heat in oil charge, Btu.
H'_T = sensible heat in storage tank, Btu.
H_p = pump energy plus or minus gains or losses from ambient between storage tank and heat exchanger, Btu.
H'_x = sensible heat in the heat exchanger, Btu.

percent utilization would indicate solids in the starting melt, no heptahydrate metastability and no solid state conversion.

The utilization values obtained could be explained by mechanisms proposed above. Thus, low salt utilization yields were probably due to crystallization of heptahydrate as a result of seeding by heptahydrate crystals formed during the preceding melting cycle. High yields resulted when a melt was successfully cooled to the temperature of dodecahydrate stability without crystallizing the heptahydrate. This demonstrates that the heat storage material must have a congruent melting point in order that its heat of fusion can be reliably realized.

Heat transfer rates were greatly improved over those reported for systems in which the storage salt was enclosed in sealed containers[3]. The oil flow rate varied from about 3 to 4 lb per min, and the temperature change in the oil across the storage container was 10 to 15 deg during that part of the freezing cycle during which latent heat was being transferred. Therefore, the minimum rate of heat transfer was about 800 Btu per hr. The excellent heat transfer was due to the high coefficients of heat transfer characteristic of direct contact heat exchange systems.

The oil bubble size was about ¼ in. in diam. Assuming no coalescence, a residence time of 1 sec and an average oil rate of 3.5 lb per min (surface area of 0.32 sq ft), an overall heat transfer coefficient of 125 Btu per (hr)(sq ft)(F deg) was calculated for the minimum heat transfer rate.

The salt volume was 0.45 cu ft, and the oil volume was 0.43 cu ft. No attempt was made to minimize the oil volume; however, the minimum amount of oil required is that which will prevent picking up salt phase in the pump suction. Based

on salt volume alone, the minimum latent heat realized on salt freezing was 4650 Btu per cu ft (cycle 74 F, Fig. 4), and the maximum was 10,200 Btu per cu ft (cycle 67 F, Fig. 5). Sensible heat in the salt is not included in these values.

CONCLUSIONS

1. The performance of a dynamic heat storage technique using the heat of fusion of a storage material has been demonstrated.

2. Subcooling of the heat storage melt was virtually eliminated as a result of the agitation produced when a nonsoluble heat exchange oil was bubbled through the storage phase.

3. Excellent rates of heat transfer were obtained as a result of direct contact heat exchange.

4. The complex crystallization mechanism of the aqueous disodium phosphate system resulted in large variations in the amount of heat able to be stored and released.

APPENDIX

CRITICAL DATA FOR HEAT STORAGE SYSTEM

1 *Storage Tank*

Material	Glass
Weight	15 lb
Exterior surface	6.3 sq ft
Heat transmission coefficient	0.1 Btu per (hr) (sq ft) (F deg)

2 *Heat Exchanger*

Concentric copper coil
Length—50 ft; 3/8-in. OD within 3/4-in. OD
Weight—21 lb
CP—0.1
Exterior surface—6.3 sq ft
Heat transmission coefficient—0.1 Btu per (hr) (sq ft) (F deg).

3 *Heat Exchange Oil*

Paraffin base mineral oil
Viscosity—32 Saybolt sec at 100 F
Sp gr—0.85
CP—0.46

4 *Storage Salt*

Aqueous Na_2HPO_4
Weight—40.2 lb
Composition—$Na_2HPO_4.12H_2O$
CP (solid)—0.37
CP (liquid)—0.65
Sp gr—1.42
Heat of fusion—($Na_2HPO_4.7H_2O$), 58 Btu per lb
($Na_2HPO_4.12H_2O$), 118 Btu per lb

REFERENCES

1. Solar House Heating—A Problem of Heat Storage, by Maria Telkes (*Heating and Ventilating*, May 1947, pp. 68–75).

2. Chemical Heat Storage for Heat Pumps by the joint committee of the *Association of Edison Illuminating Companies* and *Edison Electric Institute* (*Refrigerating Engineering*, Vol. 60, July 1952, p. 719).

3. Heat Storage Using Disodium Phosphate Dodekahydrate, by E. R. Ambrose (American Gas and Electric Service Corp. Confidential Report, January 1952).

4. Heat Storage Systems, by J. P. Howe and R. Katucki (General Electric Co. Research Laboratory Memo Report P-7, 1952).

5. Chemical Heat Storage for Heat Pumps (*Edison Electric Institute* Bulletin, September 1950, pp. 333–335).

6. Circulation in Liquid Drops, by P. H. Calderbank and I. J. O. Korchinski (*Chemical Engineering Science*, November 1956, pp. 65–78).

7. Mechanism of Heat Transfer to Liquid Drops, by R. V. McDowell and J. E. Myers (*AIChE Journal*, September 1956, pp. 384–388).

8. *Process Heat Transfer*, by D. Q. Kern (McGraw-Hill Book Co., Inc., New York, 1950, p. 563).

9. A Review of Solar House Heating, by Maria Telkes (*Heating and Ventilating*, July 1949, pp. 68–74).

10. *International Critical Tables of the Numerical Data of Physics, Chemistry and Technology*, Vols. I to VII, published under the direction of the *National Academy of Science* and the *National Research Council* (McGraw-Hill Book Co., Inc., New York, 1926–1930).

DISCUSSION

M. L. GHAI*, Cincinnati, Ohio: Referring to the reproducibility of the test data for the different tests, I am wondering if repeatability is better for the tests where the percentage of salt was less than 30 percent?

MR. ETHERINGTON: Yes.

AUTHORS' CLOSURE: I believe that the *dynamic heat storage technique* can be useful for systems using the heat of fusion of salt hydrates, provided the system is free of metastable phases.

The data indicate that the heat of fusion can be reproducibly realized, even with the disodium phosphate dodecahydrate, if the salt melt is heated long enough to remove any heptahydrate before cooling. On the other hand, I believe that successful commercial application of the disodium phosphate-water system is questionable because of the difficulty in eliminating all heptahydrate from the storage melt.

* Rocket Engine Section, General Electric Co.

1635

No. 1635

IMPROVING ATTIC SPACE INSULATING VALUES

By F. A. Joy*, University Park, Pa.

A BLACK roof top on a clear summer day can be expected to reach 150 F in any part of the United States[1]. The resulting peak heat flow into the house through the attic may determine the required capacity of air-conditioning equipment. To limit this heat load, attic insulation is an obvious need and as much as 6 in. of fibrous insulation has sometimes been recommended. Normally this is placed in the ceiling. Above it, the attic air space provides some insulating value and this value can be greatly increased by two devices: heat reflection and ventilation.

Purpose

Use of highly reflective sheet materials such as aluminum foil has long been recognized as a means of reducing radiant heat transfer but the special merit of such a sheet applied on the top of fibrous insulation in a ceiling attracted new attention following the publication of work done at the National Bureau of Standards in 1954[2]. That work was limited to simple air spaces not over 3½ in. across without joists or ventilation. For the deeper and more complex space provided by an attic and for the effect of ventilation, further research was necessary. This paper reports the results of tests on two attics, one with a flat roof and the other with a gable roof, each having 2 in. of fibrous insulation in the ceiling topped by aluminum foil, by reflective paper, and by kraft paper, with a wide range of attic ventilation.

Basic Methods

To secure realistic data, generally applicable to attic problems, the following basic methods were adopted.

1. The work was done in the laboratory under controlled, steady-state conditions rather than the transient conditions of weather exposure.

2. The attic was as large as feasible, and the perimeter heat losses were eliminated so that the results are applicable to an attic of large size.

---

* Professor of Engineering Research, The Pennsylvania State University. Member of ASHAE.

[1] Exponent numerals refer to References.

Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, January 1958.

3. The roofing temperature was 150 F, representative of full-sun summer conditions.

4. Ventilation at a controlled temperature was steady and metered. Air flow under the flat roof was parallel to the roof and ceiling joists, entering and leaving through slots at a low level. Air flow under the gable roof was across the roof and ceiling joists, entering and leaving through ports high in the gables.

5. The measured heat flow was the overall heat flow actually entering the living space through the ceiling, including the area under joists as well as that under insulation. Large heat-flow meters were made for this purpose.

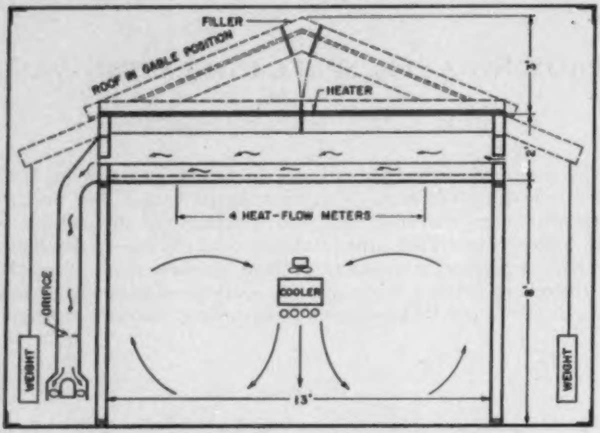


Fig. 1—Test House in Controlled Room

6. The breather sheets, kraft, foil and reflective paper, placed on the top of the fibrous insulation, were exchangeable without moving this insulation.

Test Equipment

The Test House (Fig. 1) is 12 x 13 ft inside with 8-ft ceiling height, built within a room 19 ft square with 14-ft ceiling. With a hinged and counter-balanced top, this house provides either a flat or a gable roof. All joints are gasketed for air tightness.

The roof deck is ¾-in. plywood on 2 x 6 fir rafters placed on 16-in. centers. Completely covering the deck are 2 layers of ⅛-in. cement-asbestos board, the top layer consisting of 6 sheets wound with heater wire closely spaced to provide uniform heating. These heaters are covered by 4 in. of glass fiber semi-rigid board-type insulation in 2 layers to restrict and equalize upward heat loss over the roof area. (These losses are not a part of the measurement, however.)

Each half of the deck has separate temperature control. The voltage on each heater is manually adjusted 5 percent higher than required and a 10 percent voltage correction is under automatic control. Each temperature controller uses a resistance element of fine nickel wire wound in a 6 sq ft grid on top of the plywood

deck in the central area. Close control is readily maintained since the test house is built in a controlled temperature room.

To minimize edge effects and thus insure data applicable to a much larger house, all vertical walls of the attic are *guarded* by heaters near their interior surfaces. Closely applied to the heaters and separating them from the attic space is a ½-in. thickness of insulating board faced with aluminum foil. Thermocouples on opposite sides of this board are substantially equalized by manual adjustment of the heaters.

The ceiling has 2 x 6 joists (16-in. spacing) on which 2 layers of ⅜-in. gypsum board were applied after a sheet of vapor barrier paper was stapled in place. Inlaid in the top layer are 4 heat-flow meters which are described hereafter. The bottom layer is primarily a device to shield the meters from air drafts and to insure that the ceiling is substantially airtight. The thermal value of the 2 sheets of gypsum board is comparable to a typical plastered ceiling with lightweight aggregate.

Fitted between the ceiling joists in contact with the vapor barrier are 2-in. sheets of semi-rigid glass fiber insulation having 9 lb per cu ft density. This insulation, while not typical in this application, was chosen because of its uniform thermal value and its resistance to change by accidental crushing during the progress of the work. On top of this insulation the breather sheet is placed, the following types being tested:

1. Kraft paper; 9 lb per 500 sq ft.
2. Perforated aluminum foil with kraft backing (breather type, having 64 perforations per sq in. and permeance of 40–60 perms) installed with the foil side up.
3. Reflective paper; emissivity = 0.2.

Within the test house, a brine cooled heat exchanger is placed in the center of the space. Air is blown downward by a 12-in. fan through the exchanger and immediately tempered by a bank of heating lamps. The air entrance and leaving temperatures are both measured by grids of resistance wire which act through an electronic controller to adjust the heater voltage and maintain a steady room temperature of 75 F (at mid-height). These arrangements provide low air velocity not exceeding 20 fpm 1 ft below the ceiling. A continuous record of the room temperature provided by a hygrothermograph is supplementary to thermocouple measurements taken at mid-height of the room.

Outside the test house, the temperature is controlled for general steadiness of operation and for ventilation air control.

Ventilation is provided by an exhaust fan of ample capacity. For the flat roof, air enters the attic through a 1½-in. slot running the full 12-ft width of the attic just above the ceiling joists. A cowl over the opening to obtain a more uniform inlet temperature is provided. At the outlet from the attic, the ventilation air passes through a ⅜-in. accurately adjusted slot to insure uniform air movement over the whole attic area. From the plenum beyond the slot, it passes through a straight pipe to the measuring orifice and thence to the exhaust fan.

For the gable roof, the air enters through a typical commercial louver installed in the peak after passing up through a large duct to equalize its temperature. It leaves through a horizontal ¾-in. slot 36 in. long near the peak of the opposite gable. The air flow is across the joists. The heat-flow meters were in the same positions in both flat roof and gable roof tests.

Temperature Measurements: The inlet temperature of ventilating air is obtained by 3 thermocouples placed ahead of the entrance. The outlet temperature is ob-

tained by 3 thermocouples mounted in sturdy air foils of bright aluminum mounted within and across the outlet approach where they serve to average the temperature of the outflowing air that is likely to be stratified.

Thermocouples in at least 100 locations provide a comprehensive knowledge of temperatures and operating conditions. All temperatures are recorded by a self-balancing potentiometer. A constant check of these recordings is provided by one thermocouple in ice. The locations of thermocouples in primary planes are shown in Fig. 2. In addition, an array is provided in the attic space to indicate stratification and air flow patterns. Four thermocouples are inserted into the ceiling joists at the breather level for detailed analysis of the heat distribution between insulation and joists.

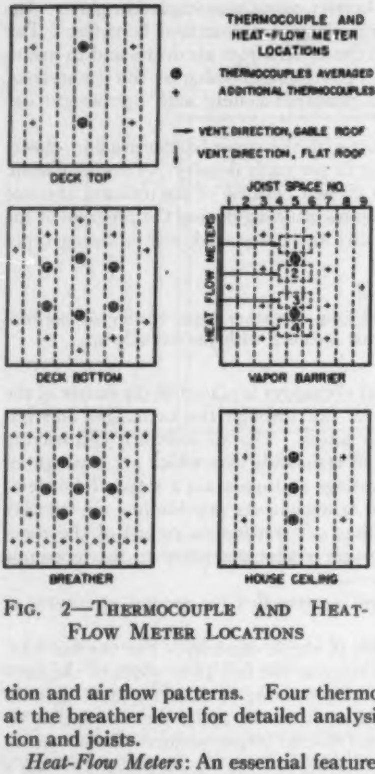


FIG. 2—THERMOCOUPLE AND HEAT-FLOW METER LOCATIONS

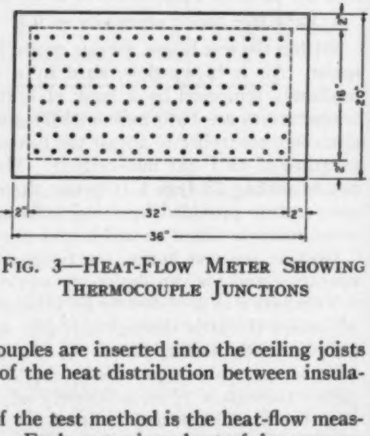


FIG. 3—HEAT-FLOW METER SHOWING THERMOCOUPLE JUNCTIONS

Heat-Flow Meters: An essential feature of the test method is the heat-flow measurement. Four heat-flow meters are used. Each meter is a sheet of dry gypsum board 20 x 36 x ⅜-in. thick on which a thermopile system containing 128 pairs of junctions is wound to cover the central area, 16 x 32 in., as shown in Fig. 3. The gypsum board was first oven dried and then painted with 2 coats of a vapor resisting paint. Copper-constantan thermocouples using #30 wire were wound in the positions shown with all wires carried around the edges—not through the meter. The meter was again oven dried and then sealed in 10-mil polyethylene adhered to the surfaces with a double faced pressure sensitive film.

As placed in the ceiling, (Fig. 2), the long dimension of a meter spans the middle (fifth) joist-space with the metered area extending from the middle of the fourth to the middle of the sixth joist-space, insuring a true composite of the total heat flow pattern under joists and insulation. The total metering area of the 4 meters is 14.2 sq ft or 9 percent of the ceiling area. The arrangement shown in Fig. 2, in the line of air flow under the flat roof, provides a good average for the ceiling area. The emf of each meter is obtained by potentiometer, manually operated.

The heat-flow meters were calibrated in a large hot plate having 32 sq ft of heater area. With the plate horizontal, 2 meters were calibrated at the same time in a 2-step procedure. One meter was placed above and the other below the heater, each separated therefrom by a sheet of high density semi-rigid glass fiber board and covered on its cold side by gypsum board. These contact materials are the

TABLE 1—HEAT-FLOW METER FACTORS

METER NO.	1	2	3	4
METER FACTOR AT 86 F	0.6720	0.6225	0.6502	0.6122
CALIBRATION TESTS NO.	C AND D		A AND B	

GLASS FIBER BOARDS k-VALUE (FOUND WITH ABOVE METER FACTORS)

	TEST NO.	C	D	A	B
Top board at 112 F Mean	Meter No. k-value	1 0.271	2 0.273	3 0.269	4 0.273
Bottom board at 113 F Mean	Meter No. k-value	2 0.265	1 0.266	4 0.263	3 0.266

same as in the test house, thus avoiding a metering question that is sometimes troublesome. The a-c heater power was measured with a watt-hour standard meter and the division of heat between the 2 meters was determined by reversing their position in the second step, assuming that the ratio of conductances of the upper and lower glass fiber boards was the same in each test. The conductivity of each board was not required though it was incidentally determined and as it provides a check of the calibration, it is reported with the meter factors in Table 1.

TEST PROCEDURE

In operation, the controlled conditions are maintained constant and the heat-flow meters are read periodically while a steady state is being established. This may require one day or a week. Moisture effects slow up the operation. To minimize these effects, the roof is continuously heated. When heat-flows and temperatures taken at 20 min intervals over a 6-hr period show no drift, the test is considered satisfactory.

TEST DATA

A summary of the results of all flat roof tests is presented in Table 2. The summary of gable roof tests is presented in Table 3. In these tabulations, the

TABLE 2—SUMMARY OF FLAT ROOF TESTS

Test No.	Breather	Ventilation: Air In F	Ventilation: Air Out F	Ventilation: Air Flow cfm	Ventilation: Rate cfm per sq ft	Center Zone Average Temperatures, Fahrenheit: Roof Top	Deck Bottom	Breather	Vapor Barrier	Ceiling	Room	Heat Flow Btu/sq ft, hr: Meter No. 1	Meter No. 2	Meter No. 3	Meter No. 4	Avg	Cor. to Std.,[a] Btu	Attic Effect. Res.,[b] Ru
1	Kraft	—	—	0	0	149.9	140.9	133.1	89.6	83.8	74.9	5.903	5.997	5.951	5.992	5.961	5.96	2.6
2	Kraft	84.6	121.9	31	0.20	149.7	134.7	122.1	87.6	82.8	76.5	4.430	—	4.870	5.153	(4.776)	4.94	5.2
3	Kraft	85.8	113.4	72	0.46	150.2	132.7	116.4	86.7	82.5	75.7	3.725	4.029	4.234	4.412	4.100	4.12	8.2
4	Kraft	85.4	101.5	162	1.04	149.6	128.6	110.3	85.6	82.0	75.9	3.067	3.309	3.414	—	(3.310)	3.38	12.2
5	Kraft	94.9	108.1	162	1.04	149.9	131.9	116.0	87.8	83.8	74.9	3.565	3.808	3.826	3.910	3.777	3.78	9.8
6	Kraft	104.0	126.9	31	0.20	151.2	138.5	127.7	89.1	84.0	76.8	5.153	—	5.417	5.690	(5.386)	5.53	3.6
7	Kraft	105.2	115.3	162	1.04	150.8	135.1	121.4	88.8	83.8	75.5	4.210	4.432	4.422	4.479	4.436	4.44	6.9
8	Perforated[c] Alum. Foil	—	—	0	0	149.6	142.4	113.3	85.9	82.0	75.3	4.224	4.336	4.275	4.359	4.299	4.34	7.3
9	Perforated Alum. Foil	86.0	119.3	31	0.20	150.6	138/9	101.6	83.9	81.7	76.8	2.594	2.859	2.959	3.125	2.884	2.94	15.5
10	Perforated Alum. Foil	85.4	106.4	72	0.46	149.9	135.7	94.1	81.8	80.0	75.3	1.843	2.109	2.236	2.323	2.128	2.13	25.2
11	Perforated Alum. Foil	84.3	95.8	162	1.04	149.9	133.9	91.2	82.0	81.1	74.7	1.512	1.777	1.791	1.811	1.722	1.75	32.8
12	Perforated Alum. Foil	95.1	105.3	162	1.04	150.8	137.5	98.5	84.1	83.1	75.7	2.237	2.480	2.437	2.449	2.410	2.45	20.6
13	Perforated Alum. Foil	104.1	123.7	31	0.20	149.8	140.1	107.6	84.7	82.3	76.4	3.449	3.615	3.594	3.707	3.591	3.74	10.0
14	Perforated Alum. Foil	105.2	117.4	72	0.46	149.8	138.6	106.8	84.1	80.9	74.4	3.398	3.522	3.495	3.539	3.488	3.43	11.8
15	Perforated Alum. Foil	105.1	113.0	162	1.04	150.6	138.9	106.5	85.2	82.7	74.3	3.217	3.386	3.294	3.295	3.298	3.22	13.3
16	Reflective Paper	84.5	110.3	72	0.46	151.1	135.3	106.1	84.1	81.5	76.2	2.871	3.154	3.272	3.476	3.193	3.29	12.8
	WINTER CONDITIONS																	
31	Perforated Alum. Foil	—	—	0	0	7.8	13.5	27.5	67.4	71.3	75.0	5.025	5.040	5.080	5.106	5.063	5.65	3.3
32	Alum. Foil	−2.0	5.3	87	0.56	1.4	3.0	11.3	65.0	70.0	74.5	6.392	6.456	6.444	6.457	6.437	6.46	1.6

[a] Corrected to standard conditions. [b] Attic effective resistance, Ru. [c] Perforated for breather application.

TABLE 3—SUMMARY OF GABLE ROOF TESTS

Test No.	Breather	Ventilation				Center Zone Average Temperatures, Fahrenheit						Heat Flow Btu/sq ft, hr					Cor. to Std.,[a] Btu	Attic Effect. Res.,[b] Ru
		Air in F	Air out F	Air Flow cfm	Rate cfm per sq ft	Roof Top	Deck Bottom	Breather	Vapor Barrier	Ceiling	Room	Meter No. 1	Meter No. 2	Meter No. 3	Meter No 4	Avg		
17	Kraft	—	—	0	0	149.8	141.8	133.5	89.0	83.8	75.5	6.294	6.375	6.488	6.371	6.382	6.44	1.6
18	Kraft	86.5	129.0	31	0.20	149.9	137.1	126.3	86.6	81.4	74.7	5.518	5.730	5.814	5.537	5.650	5.60	3.4
19	Kraft	84.7	120.4	72	0.46	150.4	132.4	119.1	85.8	82.0	75.8	4.687	4.745	4.765	4.630	4.707	4.76	5.7
20	Kraft	84.5	106.0	162	1.04	149.8	127.2	112.2	84.7	81.5	75.8	3.793	3.852	3.882	3.782	3.827	3.93	9.1
21	Kraft	106.8	130.9	31	0.20	150.2	138.9	129.1	88.7	83.9	75.6	5.802	5.862	5.924	5.797	5.846	5.86	2.8
22	Kraft	105.4	126.3	72	0.46	150.1	135.8	125.0	87.2	82.7	75.0	5.377	5.412	5.490	5.399	5.420	5.40	3.9
23	Kraft	106.1	120.2	162	1.04	150.8	132.6	121.0	86.9	83.3	76.3	4.719	4.786	4.861	4.715	4.770	4.80	5.6
24	Perforated[c] Alum. Foil	—	—	0	0	149.4	142.3	115.1	85.4	82.8	75.7	4.373	4.588	4.616	4.611	4.547	4.63	6.2
25	Perforated Alum. Foil	86.4	132.6	31	0.20	150.9	139.1	109.5	84.1	81.5	75.4	3.731	3.853	3.787	3.951	3.830	3.79	9.7
26	Perforated Alum. Foil	84.2	123.2	72	0.46	150.3	134.3	107.6	83.9	80.6	75.2	3.232	3.539	3.533	3.573	3.469	3.49	11.5
27	Perforated Alum. Foil	34.7	109.9	162	1.04	148.8	128.6	99.3	81.3	78.6	74.5	2.545	2.883	2.982	2.737	2.787	2.80	16.8
28	Perforated Alum. Foil	106.3	130.0	31	0.20	149.9	139.4	110.8	83.6	80.4	75.2	4.123	4.248	4.276	4.134	4.195	4.19	7.9
29	Perforated Alum. Foil	105.9	118.8	162	1.04	150.5	133.7	105.9	83.6	80.8	75.3	3.124	3.436	3.752	3.423	3.434	3.41	12.0
30	Perforated Alum. Foil	105.8	118.8	162	1.04	150.2	133.5	105.9	82.8	80.3	75.3	3.125	3.424	3.762	3.454	3.441	3.43	11.9
	WINTER CONDITIONS																	
33	Perforated Alum. Foil	—	—	0	0	5.1	10.9	25.6	66.9	71.3	74.6	5.223	5.304	5.252	5.201	5.270	5.69	3.2
34	Alum. Foil	−0.3	8.0	87	0.56	2.4	5.2	16.5	65.6	70.6	74.3	6.048	6.154	6.112	6.089	6.101	6.27	1.9

[a] Corrected to standard conditions. [b] Attic effective resistance, Ru. [c] Perforated for breather application.

average temperature at each major plane of the construction includes only those in the central zone—the circled points in Fig. 2. Outside this zone, the temperatures were not much different, but they are omitted from the average for greater refinement.

ANALYSIS OF SUMMER HEAT GAIN TESTS

The heat flow measured by each meter is listed in the tables and its position is shown in Fig. 2. After averaging all meters, the heat flow is corrected to standard conditions: 150 F roof top; 75 F room; and 85 F, 95 F or 105 F ventilating air. Evidence of good reproducibility is shown in Tests No. 29 and No. 30, not consecutively run. The corrected heat flows are plotted in Fig. 4 against ventilation rate expressed as cfm per sq ft (of ceiling area).

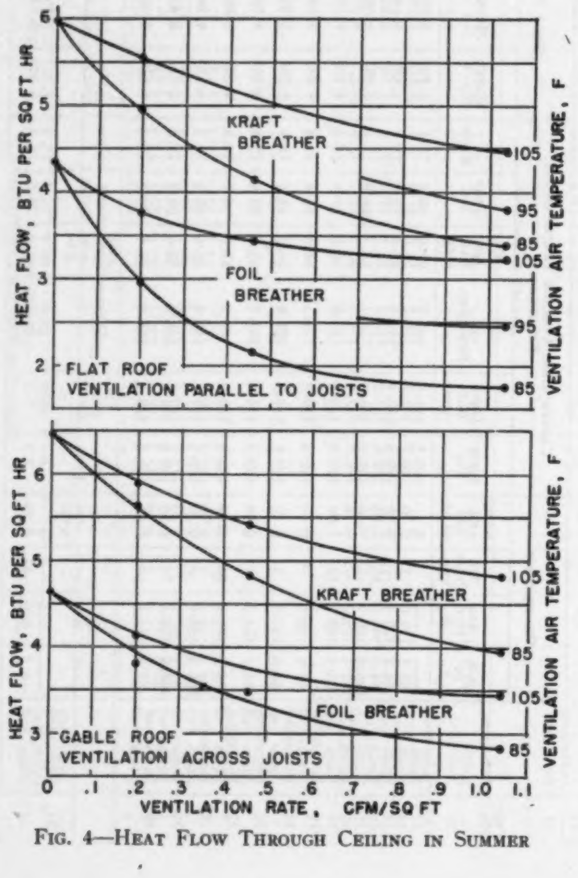


FIG. 4—HEAT FLOW THROUGH CEILING IN SUMMER

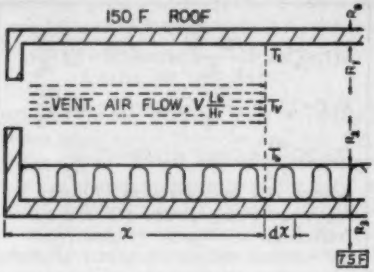


FIG. 5—ATTIC SECTION

Application of these data to a house larger than the one tested merits consideration. The simplest enlargement is imagined by placing 2 attics (like that tested) side by side with cross ventilation doubled in volume. The ventilating rate expressed in cfm per sq ft is the same, and the heat inflow obviously will be the same in the second house. Therefore, Fig. 4 applies correctly to any attic where the air path is 13 ft long.

Considering the same 2 attics ventilated the long way, in a 26-ft path, the application of Fig. 4 is less clear. However, if the temperature of the air entering the second attic is taken as it is discharged from the first, the heat flow for each is obtainable from Fig. 4 and these results may be averaged for the whole ceiling. Continuing in the same way, the results for 3 or more attics can be combined. Applying this step-by-step procedure, the data indicate that Fig. 4 is properly applicable to longer air paths.

To confirm this conclusion, a mathematical analysis is readily made if certain simplifying assumptions are allowed. Provided the surface coefficient of heat transfer on the deck bottom and breather top are constant within the range considered, the heat balance in Fig. 5 may be developed as shown in Fig. 6.

In Equation 1, it is evident that Q_{avg} is constant when V/L is chosen, demonstrating that Fig. 4 is valid for any length of air path.

The assumptions are reasonable if the entering and leaving velocities are held constant by adjusting the port size in proportion to the ceiling area which is good practice. With an attic height as tested, it appears that Fig. 4 may be used without serious error for any air path up to 40 ft long.

Exact computations for an attic of different height or roof slope cannot be made directly from the basic data. However, it is evident that the position, size and shape of the air entrance and exit ports may have a very important effect on the value of ventilating air. These factors were optimum in the flat roof design. They were very unfavorable in the gable roof.

Perimeter heat losses were carefully avoided in the tests so that the data could be applied to a larger house. If the attic wall area is large and uninsulated, the heat loss from the attic to the weather merits attention, but its net effect is not simply stated.

WINTER HEAT LOSS TESTS

To complete the program, four tests were run under winter conditions holding the temperature at zero F in the space surrounding the test house. For this pur-

$$N(T_1-T_b)dx+\left(\frac{T_1-T_v}{R_1}\right)dx=(150-T_1)\frac{dx}{R_0}$$

$$N(T_1-T_b)dx-\left(\frac{T_b-T_v}{R_2}\right)dx=(T_b-75)\frac{dx}{R_3}$$

$$(T_b-75)\frac{dx}{R_3}+CV\,dT=(150-T_1)\frac{dx}{R_0}$$

where the temperatures (T) and thermal resistances (R) are as indicated in Fig. 5. In addition:

dT=Small temperature rise (F deg) of ventilating air as it passes over the increment, dx.

C=Specific heat of air, Btu/(lb)(F deg)

N=radiation coefficient between roof bottom and the breather (approximately constant for this purpose), Btu/(sq ft)(hr)(F deg).

Eliminating T_1 and T_v from the heat balance equation, gives:

$$T_b=K_0+K_1e^{-\frac{K_2}{V}x},\quad \text{where}$$

K_0, K_1 and K_2 are constants including C, N, R_0, R_1, R_2, and R_3.

Total heat flow into the house along the full path, L, is written as:

$$Q_L=\int_0^L\left(\frac{T_b-75}{R_3}\right)dx$$

$$=\frac{1}{R_3}\int_0^L\left[\left(K_0+K_1e^{-\frac{K_2}{V}x}\right)-75\right]dx$$

Integrating and dividing by L gives the average heat flow Q_{av}, as:

$$Q_{av}=\frac{1}{R_3}\left[(K_0-75)+\frac{K_1}{K_2}\left(\frac{V}{L}\right)\left\{1-e^{-K_2\left(\frac{L}{V}\right)}\right\}\right] \text{-----}(1)$$

FIG. 6—MATHEMATICAL ANALYSIS OF HEAT BALANCE IN FIG. 5

pose, all insulation was removed from the roof top but the cement-asbestos sheets remained (unheated). The insulation in the wall areas of the attic remained but the *guard heaters* were idle. Inside the house, heating and air circulation as in the summer tests were provided.

Moderate ventilation was supplied in 2 tests. Attic ventilation is unwanted in winter, but a small amount is required to avoid condensation of moisture coming from the occupied space. For this purpose, certain vent areas have been recommended[8] with natural ventilation. The ventilation rate varies with wind but it is believed that the tested rate—0.56 cfm per sq ft—exceeds normal in most loca-

tions. The winter results are presented in Tables 2 and 3 including the heat loss corrected to standard conditions: 75 F room with 0 F roof top and ventilating air.

VALUE OF THE ATTIC

The temperatures recorded for the summer tests in Tables 2 and 3 show clearly that the breather is always much cooler than the roof top, the difference indicating the insulating value of the attic. To present this observation in organized form, it is possible to compute the thermal resistance of the attic in the usual way, the heat flow and temperature difference being known. When there is attic ventilation, the result may be called *effective resistance* though its application is the same.

Actually, the breather temperature should be modified to include the effect of the ceiling joists which have a different temperature and are indeed an important heat path by-passing the insulation when it is covered by aluminum foil but the joists are not. An expedient is to calculate the U-value for the ceiling, including joists, by accepted methods[4] and then subtract its resistance $(1/U)$ from the tested resistance of the structure. The remainder is the resistance in the heat path from roof top to the *effective top* of the ceiling unit in air near the breather.

For the ceiling unit here reported, the calculation of the U-value (using test data where available) is as follows:

	RESISTANCE
Insulation: 1.98 in. thick, $k = 0.265$	7.48 ru[a]
Two sheets gypsum board	0.64
Surface above (still air)	0.92
Air below (from test data)	1.50
Total	10.54 ru[a]

[a] *Resistance unit*, abbreviated to ru, is 1 (hr) (sq ft) (F deg) per Btu and is here proposed as a convenient name for this unit since the longer description is awkward. Being awkward and lacking physical significance to many people, the unit is unnamed in a recent publication though resistance and conductance are its subject[4]. The short name is used herein.

The usual correction for joists treats them as parallel heat-flow paths. The observed temperature within joists at the breather level averaged 10.4 F lower than the breather temperature in Tests No. 1 and No. 17, without ventilation. Consequently, the depth of joist to be considered in this assembly is 2.8 in. (the depth of insulation plus one half the joist width). On this basis, the U-value correction for joists is 1.06 and the calculated ceiling resistance $(1/U)$ with non-reflective breather is 10.0 ru.

The effective resistance of the attic in each test is found by subtracting 10.0 ru from the overall resistance of the structure since the ceiling was always the same except the breather. Any effect of a highly reflective breather is properly credited to the air space above it and not to the elements it touches. The procedure outlined gives the credit where it belongs—to the attic. The effective resistance of the attic is shown in Tables 2 and 3 and in Figs. 7 and 8.

For the winter tests, the attic resistance is 3.3 ru under the flat roof and 3.2 ru under the gable roof, both having aluminum foil breathers and no ventilation. The breather value is less in winter with upward heat flow. Ventilation lowers the effective resistance and should be held to proper limits. These winter results are included in the tables.

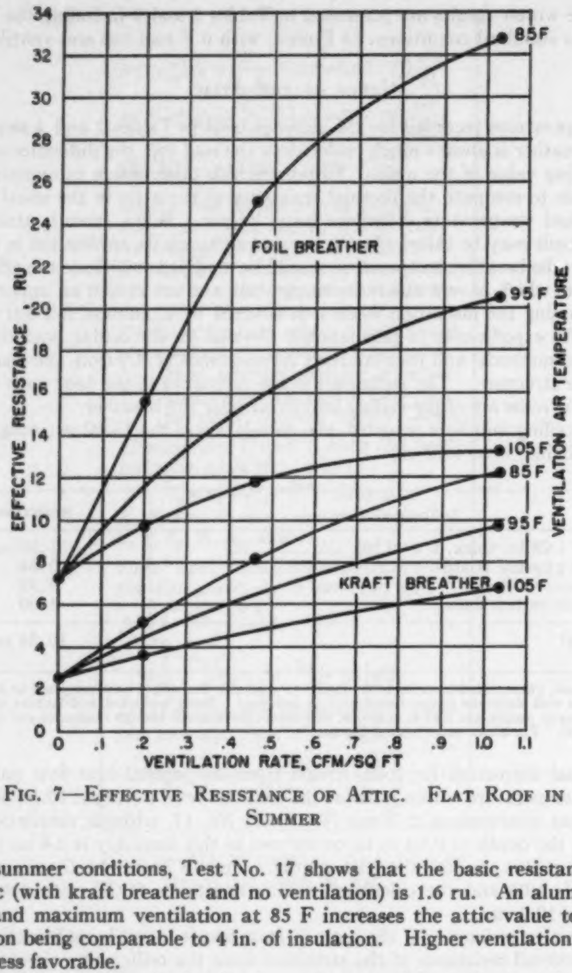


FIG. 7—EFFECTIVE RESISTANCE OF ATTIC. FLAT ROOF IN SUMMER

Under summer conditions, Test No. 17 shows that the basic resistance of the gable attic (with kraft breather and no ventilation) is 1.6 ru. An aluminum foil breather and maximum ventilation at 85 F increases the attic value to 16.8 ru, the addition being comparable to 4 in. of insulation. Higher ventilation temperatures are less favorable.

With the flat roof (Fig. 7) the effective resistance of the attic rises sharply with 85 F ventilation, especially when the breather is aluminum foil. This appears to be due to laminar flow of the ventilating air with a high degree of stratification in the attic space. This result is accomplished by placing the inlet and outlet low (just above the ceiling joists) with low entering velocity through a slot extending the full width of the ceiling. It may be noted that even more attic resistance could be obtained by covering the exposed joist tops with reflective material.

Under all conditions, the gable attic has lower resistance than the flat roof design. With no ventilation this difference is traceable primarily to the reduced effect of

joists blocking direct radiation. Convection of air is also encouraged by the wedge shape of the space and the roof surface per square foot of ceiling is increased. But the lower effectiveness of ventilation is due primarily to the air path which is distinctly unfavorable in the gable attic.

With the practical problems of design and construction it may not be feasible to secure all the added insulating values shown in the flat roof tests, but it surely is feasible to attain those in the gable tests.

ACKNOWLEDGMENTS

The work here reported was done in The Thermal Research Laboratory at The Pennsylvania State University under the sponsorship of the Aluminum Company of America. Mr. J. J. Zaborny did much of the test work with important contributions by other members of the laboratory staff.

REFERENCES

1. Principles Affecting Insulated Built-up Roofs, by C. E. Lund and R. M. Granum (*Engineering Experiment Station Bulletin No.* 34, University of Minnesota, Minneapolis, Minn.).
2. The Thermal Insulative Value of Air-spaces, Housing Research Paper 32 (U. S. Housing and Home Finance Agency, Washington, D. C.).
3. Condensation Control in Dwelling Construction (U. S. Housing and Home Finance Agency, 1950).
4. HEATING VENTILATING AND AIR CONDITIONING GUIDE, 1957, Chapter 9—Heat Transmission Coefficients of Building Materials (published by AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.)

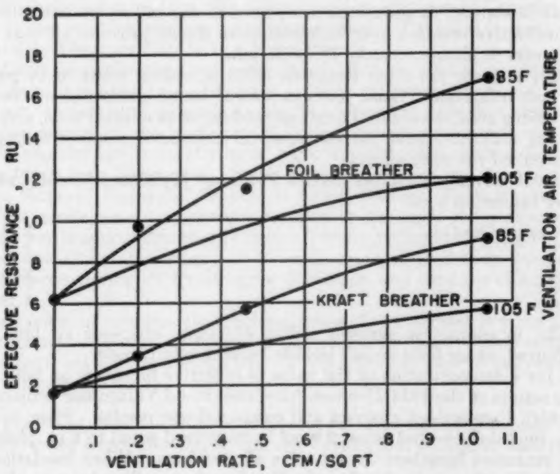


FIG. 8—EFFECTIVE RESISTANCE OF ATTIC. GABLE ROOF IN SUMMER

5. Heat Loss Calculations, Technical Circular No. 7, Revised April 1956 (U. S. Federal Housing Administration, Washington, D. C.).

DISCUSSION

R. H. HEILMAN, Pittsburgh, Penna., (WRITTEN): Professor Joy is to be commended for the excellent paper he has presented and for the apparent accuracy with which the tests were conducted.

I have no criticism of the results obtained, but I feel that one statement he has made, namely, under summer conditions, *An aluminum foil breather and maximum ventilation at 85 F increases the attic value to 16.8 ru, the addition being comparable to 4 in. of insulation*, may cause some hesitancy on the use of conventional insulation.

The occupant of the residence using forced ventilation would have to pay for this ventilation every time it is turned on while the installed insulation would present to the owner a return on his original investment every heating season that the house is occupied. According to Table 3, the resistance of maximum ventilation alone at 85 F is 9.1 − 1.6 = 7.5 ru comparable to 2 in. of insulation, so that the resistance of the perforated aluminum foil is comparable to only 2 in. of insulation for summer conditions only, and only when the foil retains its original highly reflecting surface.

We are constantly being called by people who have been misinformed on the use of insulation. Only a few days ago, a distraught lady called who had bought full-thick-bat insulation with vapor barrier for the walls of her new home. After she obtained the insulation, she was advised by two building contractors not to use it, for if she installed it without an air space, she was informed that the studding would soon rot out.

I know Professor Joy's statement is accurate and would cause no misunderstanding among engineers reading this paper, but from past experiences, I feel that it could be used to unfair advantage on the uninformed public.

J. R. WATT, Austin, Texas, (WRITTEN): Professor Joy has greatly contributed to insulating and building knowledge. We hope his data destroys the common superstition that roof heat in summer is convected downward to the ceiling by hot attic air. His success with reflective breather over bulk insulation should prove to all that downward attic heat transfer is almost entirely by radiation.

He rightfully stresses the often forgotten effect of ceiling joists in by-passing heat around and past insulation. Since joists in normal frame construction occupy 10 percent of total ceiling area, are about 3 times as conductive as mineral wool, and have large areas absorbing radiation from the hot roof, all calculated summer ceiling *U*-values should be corrected for joist effect.

My calculated correction factors exceed Professor Joy's considerably, varying also with depth of insulation used:

2 in. mineral wool	1.14
3 in. " "	1.15
4 in. " "	1.16
5 in. " "	1.165
6 in. " "	1.17
8 in. " "	1.18

Incidentally, in computing summer ceiling *U*-values, attic-side air film coefficients should be omitted, as air films do not impede radiant heat transfer.

Professor Joy's demonstration of the value of reflective breathers on bulk insulation, corroborates results of the 1954 22-house Air-Conditioned Village test project of Austin, Texas, for which I supervised analyses and computed test results. Here, in actual residences, 4 in. thick foil-covered mineral wool batts proved equal to 6 in. plain ones.

However, reflective breathers do not solve all problems. When insulation thickness is significantly less than joist depth, reflective breathers reflect heat on the joists' exposed flanks, intensifying their effect. Thus, best practice should use insulation of full joist depth, burying all but their top edges against radiation.

It is hoped that manufacturers will soon provide further solution in prepared roof decking with reflective undersurface.

Professor Joy does great service in delineating the possibilities of forced attic ventilation. The Austin Air-Conditioned Village had 3 residences with 1/6 hp exhaust fans drawing air through gabled attics. No benefit was measurable, save cooler attic air. This intensified the fear of many of us that only heroic air flows could significantly assist insulation in gabled attics. Analytically, the problem is as follows.

The Paradox of Attic Ventilation

1. Any ventilation which reduces the temperature of the roof's underside increases heat conduction down through it.

2. Any ventilation which reduces the temperature of the joists and ceiling insulation increases radiant flow to them from the hot areas above.

3. Thus, removing heat from these surfaces induces compensatory heat flows counteracting the ventilation effect.

4. Consequently, only air flows removing heat *faster than it can be replaced* can significantly aid attic insulation.

Professor Joy's success with low air flow rates apparently disproves this thesis. His ventilating rates from 0.2 to 1.04 cfm per sq ft of ceiling approximate only 0.15 to 0.70 air changes per minute in modern low-gabled attics, with velocities not exceeding 2 fpm.

It is curious that such modest ventilation can succeed. While the entering air's density helps spread it over the attic floor, the transverse joist edges form wind-breaks sheltering much insulation from it, so contact seemingly is poor.

We hope Professor Joy continues his careful work and enlightens us further.

H. E. ROBINSON, Washington†, D. C.: I would like to make two comments in connection with this excellent paper by Professor Joy.

The difference of temperature of the roof surface and the entering attic ventilating air is of major importance in evaluating the *effective resistance* of the attic, as shown in Figs. 7 and 8. Analysis indicates that this difference is of considerably greater importance than the attic temperature level, or mean temperature. Thus it is important that it be known what this difference may be in a given case.

As a good first approximation, and assuming the extreme case of no outside wind, the rise of roof surface temperature above the outdoor air temperature (*i.e.*, the entering ventilating air temperature) depends on the incident solar radiation intensity, and on the solar absorptance of the roofing and its low temperature emittance. For nonmetallic surfaces, the last is usually quite high, but the solar absorptance depends on color, white being least absorptive. Some comparative data on this point have been published in National Bureau of Standards *Report* BMS64, *Solar Heating of Various Surfaces*. The maximum solar intensity depends on sun angle and atmospheric clarity, but for wide areas is approximately the same. Hence, it should be possible to establish, for representative kinds of roofing, and for wide areas, approximate values of the extreme roof surface-entering air temperature difference, and thus the effective resistance of the attic.

My second comment concerns the felicitous name that Professor Joy has proposed for the unit of thermal resistance. I hope this name will find favor—some simple term for this unit has been needed for a long time. I will add only that in my opinion, if you RU it, you will not rue it.

WARREN VIESSMAN, Baltimore, Md.: Professor Joy is to be complimented on presenting a very interesting paper and also on the instrumentation technique used in obtaining his results.

† National Bureau of Standards

I would like to ask a question regarding the perforated surfaces. What is the size of the perforations and their density? I presume that they are merely provided to release moisture in order to keep the ceiling structure dry and have no appreciable effect on the heat transfer. I would like a statement in regard to this.

Another point that I would like to have clarified is that the test results seem to indicate that a flat roof might be more desirable than a gable roof, as the test results indicate a little higher transmission with a gable roof.

I think probably that is an erroneous conclusion, when all the factors such as sun effect on a gable roof are considered. The differences in the test results are probably attributed to the air flow path and construction differences, since the air flow in the two cases are at right angles and parallel to the joists.

AUTHOR'S CLOSURE: Mr. Heilman properly draws attention to the fact that ventilation is not the equivalent of insulation throughout the year. Most of my data apply to a peak roof temperature in summer, showing that ventilation can be very effective in reducing the heat inflow to the house at such times. It is also effective in the evening. A full analysis of daily temperature cycles would show the times when attic ventilation can be used with profit and it would be a valuable addition. In the meantime, I caution that no statements in the paper should be taken out of context.

Dr. Watt has good grounds for omitting the top side film resistance of the ceiling unit for many purposes. I have included it because it is conventional and is included in Table 16 of THE GUIDE. This avoids complications but sometimes gives the ceiling a credit (0.92 ru) which perhaps belongs to the attic. No error results in the combination however, since the attic resistance is always derived from the tested overall resistance. His factor (1.14) for the effect of ceiling joists with 2 in. of insulation between them applies, I think, when the joist depth is also 2 in., so that the same top side temperature is found in both areas. The assemblies tested had 2 x 6 joists but the observed temperature inside the joists at the 2-in. level was 10.4 F deg cooler than the center of the kraft breather. It may be inferred that the edges of the breather were also cooler. These temperatures reduce the *joist effect* materially as do the auxiliary resistances which cover all the area. Actually, the indicated factor from tests No. 1 and No. 17 averages nearer 1.05.

Dr. Watt's *paradox* properly suggests that the roof deck resistance is a factor in the effectiveness of ventilation and leads to the question of optimum division of insulation between ceiling and deck. His suggestion of a reflective under surface of the deck is good. I am not surprised that he was unable to see the effect of attic ventilation in the cooling load of houses at Austin. There were many complex factors there. Also, perhaps these attics had more than 2 in. of insulation, thereby reducing the net saving by ventilation.

Mr. Robinson has an interesting suggestion for correlating the roof top temperature and ventilating air temperature. As he says, this is possible for a given roofing if the sky is clear and there is no wind. This basis merits further study along with the study of off-peak conditions that I have already mentioned.

To Mr. Viessman I can answer that the foil had 64 minute perforations per sq in. They are effective in the release of water vapor and their effect on heat transfer is negligible. The larger gain by ventilation under the flat roof is mostly due to the air flow path which was low and parallel to the ceiling joints. I believe that nearly as good results could be obtained under a gable roof if it is feasible to provide a similar air path.

1636

No. 1636

CHIMNEY AND STACK DESIGN FOR GAS-FIRED EQUIPMENT

By RICHARD L. STONE*, BELMONT, CALIF.

DRAFT requirements of gas-burning equipment, including furnaces, ovens, boilers and related units, depend on whether or not the equipment has a draft hood. Those units with draft hoods need no stack action except that produced in the flue ways and within the combustion chamber of the equipment itself. Those without draft hoods may require the aid and assistance of a proper chimney or stack to burn the fuel at the desired rate to maintain desired internal pressure, to assure adequate air for correct combustion, and to sustain maximum firing rates.

The design methods for stacks chimneys, and gas vents presented in this paper resulted from a study of the theoretical relationships governing chimney and gas vent operation. This study was directed at developing methods of tabulating important factors affecting draft and capacity in terms familiar to and common in the gas industry. The equation from which these tables were computed is related to those given in the ASHAE GUIDE[1] and in other references on this subject[2, 3]. As is commonly known, the draft equation shows that available draft results from the difference between *theoretical* draft and energy losses due to flow. Common usage dictates the choice of the word *theoretical* although this effect is more correctly described as potential draft.

In a boiler-chimney system containing a draft diverter, available static draft applied at the draft diverter inlet, or to the boiler flue outlet is zero for all practical purposes because the entire system theoretical draft is balanced out by flow energy losses, which result from the draft-nullifying entrance of dilution air through the relief opening. Where the gas-burning equipment requires a fixed or variable amount of available draft, the chimney, breeching, and draft control system must be designed to produce the maximum amount needed. This draft may then be reduced by appropriate control devices to the amount needed to meet operating demands.

This approach permits the problem of chimney design for gas equipment needing draft to be separated into a 2-step procedure. These are: (1) from required

---

* Chief Research Engineer, William Wallace Co.

[1] Exponent numerals refer to References.

Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, January 1958.

draft based on expected stack gas temperature and equipment characteristics such as efficiency, over-fire static pressure, firing rate, steam pressure, or temperature, choose a stack height which will yield a theoretical draft greater than that needed for maximum available static draft. Note that other considerations such as building height, fume and gas dispersal, location of adjacent structures, may require greater chimney height than the amount theoretically called for; (2) from chimney operating conditions, choose a diameter at which the sum of the energy losses plus the available draft equals the theoretical draft.

Expressed as a simple equation this procedure states that:

Potential Static Draft = Available Static Draft + Static Draft Losses.

The exact terms of this equation are given in the Appendix, along with a complete derivation, and an illustrated example. While an understanding of the equation is helpful to its direct application in solving problems, no reference to it need be made to use the tables given here for rapid chimney and stack design.

TABLE 1—THEORETICAL OR POTENTIAL STATIC DRAFT PRESSURE PRODUCED UNDER NO-FLOW CONDITIONS IN A VENTING SYSTEM, INCHES OF WATER COLUMN

	Average Temperature Rise Above Ambient 60 F of Gases in Vertical Part of Chimney, F Deg							
Height of Chimney, Feet, Above Level of Draft Measurement	150°	200°	300°	400°	500°	600°	750°	1000°
5	0.017	0.020	0.027	0.032	0.036	0.039	0.043	0.049
10	0.033	0.041	0.054	0.064	0.072	0.079	0.087	0.097
15	0.049	0.061	0.081	0.096	0.108	0.118	0.130	0.145
20	0.066	0.082	0.11	0.13	0.14	0.16	0.17	0.19
25	0.083	0.10	0.13	0.16	0.18	0.20	0.22	0.24
30	0.10	0.12	0.16	0.19	0.22	0.24	0.26	0.29
40	0.13	0.16	0.22	0.26	0.29	0.32	0.35	0.39
60	0.20	0.25	0.32	0.38	0.43	0.47	0.52	0.58
80	0.26	0.33	0.44	0.51	0.58	0.63	0.70	0.78
100	0.33	0.41	0.54	0.64	0.72	0.79	0.87	0.97
150	0.50	0.61	0.81	0.96	1.08	1.18	1.30	1.45
200	0.66	0.82	1.07	1.28	1.44	1.58	1.74	1.94
300	0.99	1.23	1.61	1.92	2.16	2.37	2.61	2.91

STACK DESIGN METHODS

The chimney design tables, based on the draft equation, can be readily employed to calculate the height and size of stacks and chimneys for gas equipment requiring draft, as well as for appliances having draft hoods. With either type of equipment it is necessary to start with Table 1 to obtain theoretical draft solely as a function of height and gas temperature. Next, the draft losses must be found using Table 2, 3, or 4. Tables 2 and 3 are applicable to chimney systems not having draft hoods, and give draft loss due to energy losses from the flowing stack gases containing 4 percent and 8 percent CO_2 respectively. Table 4 applies to gas venting where a draft hood is used. The stack or chimney is designed for draft using the applicable energy loss table so that draft produced by the stack will be greater

than the equipment needs by a margin sufficient to insure proper equipment operation under all stack flow conditions. For gas vents, the system is designed for zero draft by equating losses from Table 4 to theoretical draft for the chosen height.

The typical arrangement of gas equipment, lateral connector, breeching, and vertical stack or chimney to which Tables 1, 2, 3, and 4 apply is shown in Fig. 1. Tables 2, 3 and 4 contain column headings of recommended L/D values for those

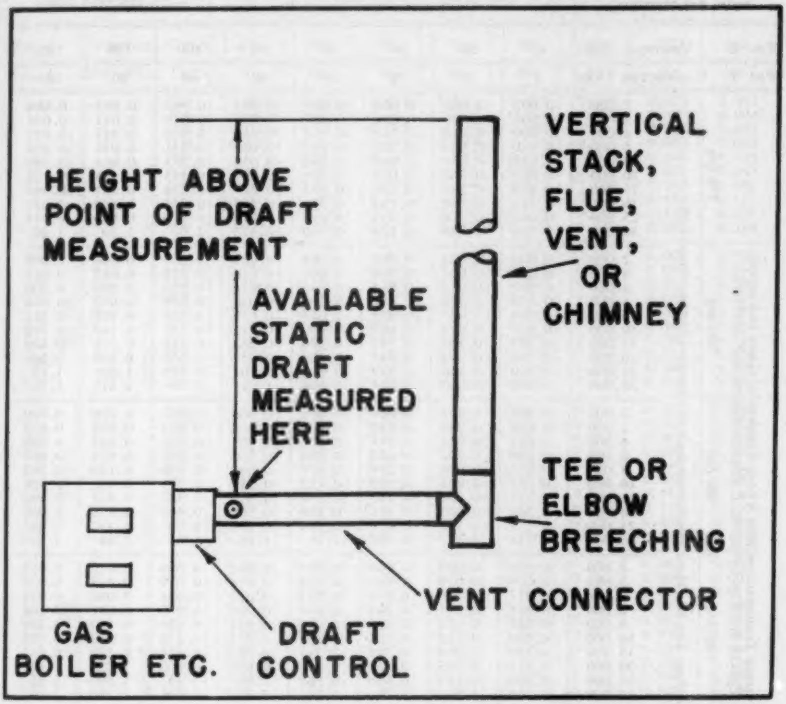


FIG. 1—LOCATION OF AVAILABLE DRAFT MEASUREMENT IN TYPICAL BOILER STACK SYSTEM

chimneys having lateral connectors (Fig. 1) as well as for vents and stacks which consist of vertical piping only and have no lateral connection (Fig. 2).

The reader should be forewarned that stack designing by application of the tables given here requires cut-and-try calculation procedures, but by doing so, cut-and-try chimney construction methods may be eliminated.

The following terms and headings in the Tables 2, 3, and 4 should be familiar to users of gas equipment. With the help of some auxiliary tables, the effects of operating variables can be estimated and the height and size of the stack or chimney can be computed with a minimum of effort.

TABLE 2—STATIC DRAFT PRESSURE LOSSES IN VENTING SYSTEM WITH 4 PERCENT CO_2 IN THE STACK GASES, INCHES OF WATER COLUMN

(See Table 4 for Energy Losses in Systems Having Draft Hoods)

FOUR PERCENT CO_2		L/D; SYSTEM LENGTH TO DIAMETER RATIO, FEET OF CONNECTOR AND STACK PER FOOT OF DIAMETER							
FIG. 2	VERTICAL ONLY	45	50	60	70	80	100	120	140
FIG. 1	CONNECTOR USED	5	10	20	30	40	60	80	100
Average Temperatures Rise Above Ambient of Gases in Vertical Part of Stack or Chimney, F Degrees Above 60 F Ambient	Input Loading: Btu Per Hour Per Square Inch of Vent or Chimney Area: I/A								
150 deg	2,000	0.002	0.002	0.003	0.003	0.004	0.005	0.005	0.006
	3,000	0.005	0.005	0.006	0.007	0.008	0.010	0.012	0.014
	4,000	0.009	0.010	0.011	0.013	0.015	0.018	0.022	0.025
	6,000	0.020	0.022	0.025	0.029	0.033	0.041	0.048	0.056
	8,000	0.035	0.039	0.045	0.052	0.059	0.073	0.086	0.100
	10,000	0.055	0.060	0.070	0.081	0.092	0.113	0.134	0.155
	15,000	0.123	0.135	0.159	0.183	0.207	0.254	0.301	0.349
	20,000	0.219	0.240	0.283	0.326	0.368	0.453	0.536	0.622
	25,000	0.343	0.376	0.443	0.510	0.576	0.710	0.840	0.975
	30,000	0.490	0.540	0.634	0.730	0.825	1.02	1.20	1.40
300 deg	2,000	0.003	0.003	0.003	0.004	0.004	0.006	0.007	0.008
	3,000	0.006	0.007	0.008	0.009	0.010	0.012	0.015	0.017
	4,000	0.011	0.012	0.014	0.016	0.018	0.022	0.026	0.030
	6,000	0.024	0.026	0.031	0.036	0.040	0.049	0.059	0.068
	8,000	0.043	0.047	0.055	0.063	0.072	0.088	0.105	0.122
	10,000	0.067	0.073	0.086	0.099	0.112	0.137	0.163	0.190
	15,000	0.151	0.165	0.195	0.224	0.253	0.311	0.379	0.428
	20,000	0.268	0.294	0.347	0.400	0.451	0.555	0.656	0.761
	25,000	0.420	0.460	0.542	0.624	0.705	0.870	1.03	1.19
	30,000	0.600	0.660	0.775	0.894	1.01	1.25	1.47	1.71
500 deg	2,000	0.003	0.004	0.004	0.005	0.006	0.007	0.008	0.010
	3,000	0.008	0.008	0.010	0.011	0.013	0.015	0.018	0.021
	4,000	0.013	0.015	0.017	0.020	0.022	0.027	0.033	0.038
	6,000	0.030	0.033	0.038	0.044	0.050	0.062	0.073	0.085
	8,000	0.055	0.058	0.069	0.079	0.089	0.110	0.130	0.152
	10,000	0.083	0.091	0.107	0.123	0.139	0.171	0.203	0.236
	15,000	0.187	0.206	0.242	0.279	0.315	0.387	0.458	0.531
	20,000	0.334	0.365	0.431	0.496	0.560	0.690	0.816	0.946
	25,000	0.522	0.572	0.675	0.776	0.877	1.08	1.28	1.48
	30,000	0.746	0.822	0.965	1.11	1.26	1.55	1.83	2.12
750 deg	2,000	0.004	0.005	0.005	0.006	0.007	0.009	0.010	0.012
	3,000	0.009	0.010	0.012	0.014	0.016	0.019	0.023	0.026
	4,000	0.017	0.018	0.021	0.024	0.028	0.034	0.040	0.047
	6,000	0.037	0.041	0.048	0.055	0.062	0.077	0.091	0.106
	8,000	0.067	0.073	0.085	0.098	0.111	0.136	0.162	0.189
	10,000	0.104	0.113	0.133	0.153	0.173	0.212	0.252	0.294
	15,000	0.233	0.256	0.302	0.347	0.382	0.481	0.570	0.661
	20,000	0.415	0.455	0.537	0.618	0.698	0.860	1.02	1.18
	25,000	0.650	0.713	0.840	0.968	1.09	1.35	1.59	1.85
	30,000	0.930	1.03	1.20	1.38	1.57	1.94	2.28	2.64
1000 deg	2,000	0.005	0.005	0.006	0.007	0.008	0.010	0.012	0.014
	3,000	0.011	0.012	0.014	0.016	0.019	0.023	0.027	0.032
	4,000	0.020	0.022	0.026	0.029	0.033	0.041	0.048	0.056
	6,000	0.045	0.049	0.057	0.066	0.074	0.092	0.109	0.127
	8,000	0.080	0.087	0.102	0.117	0.133	0.163	0.194	0.226
	10,000	0.124	0.135	0.159	0.183	0.207	0.254	0.302	0.352
	15,000	0.279	0.307	0.361	0.416	0.470	0.577	0.683	0.791
	20,000	0.497	0.545	0.642	0.740	0.835	1.03	1.22	1.41
	25,000	0.779	0.854	1.01	1.16	1.31	1.61	1.91	2.21
	30,000	1.11	1.23	1.43	1.66	1.87	2.32	2.73	3.16

TABLE 3—STATIC DRAFT PRESSURE LOSSES IN VENTING SYSTEM WITH 8 PERCENT CO_2 IN THE STACK GASES, INCHES OF WATER COLUMN

(Also Applies If Barometric or Damper Type Draft Control is Used)

EIGHT PERCENT CO_2		L/D SYSTEM LENGTH TO DIAMETER RATIO, FEET OF CONNECTOR AND STACK PER FOOT OF DIAMETER							
FIG. 2	VERTICAL ONLY	45	50	60	70	80	100	120	140
FIG. 1	CONNECTOR USED	5	10	20	30	40	60	80	100
Average Temperature Rise Above Ambient of Gases in Vertical Part of Stack or Chimney, F Degrees Above 60 F Ambient	Input Loading: Btu Per Hour Per Square Inch of Vent or Chimney Area: I/A								
150 deg	2,000	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.002
	3,000	0.001	0.002	0.002	0.002	0.002	0.003	0.003	0.004
	4,000	0.002	0.003	0.003	0.004	0.004	0.005	0.006	0.007
	6,000	0.005	0.006	0.007	0.008	0.009	0.011	0.013	0.015
	8,000	0.010	0.011	0.012	0.014	0.016	0.020	0.024	0.028
	10,000	0.015	0.017	0.019	0.022	0.025	0.031	0.037	0.043
	15,000	0.034	0.037	0.044	0.050	0.057	0.070	0.083	0.096
	20,000	0.061	0.066	0.078	0.090	0.101	0.125	0.148	0.171
	25,000	0.094	0.104	0.122	0.140	0.159	0.195	0.232	0.270
	30,000	0.136	0.149	0.175	0.201	0.227	0.280	0.332	0.384
300 deg	2,000	0.001	0.001	0.001	0.001	0.001	0.002	0.002	0.002
	3,000	0.002	0.002	0.002	0.002	0.003	0.003	0.004	0.005
	4,000	0.003	0.003	0.004	0.004	0.005	0.006	0.007	0.008
	6,000	0.007	0.007	0.009	0.010	0.011	0.014	0.016	0.019
	8,000	0.012	0.013	0.015	0.018	0.020	0.025	0.029	0.034
	10,000	0.019	0.020	0.024	0.027	0.031	0.038	0.045	0.052
	15,000	0.042	0.046	0.054	0.062	0.070	0.086	0.102	0.118
	20,000	0.074	0.081	0.096	0.110	0.124	0.153	0.182	0.209
	25,000	0.116	0.127	0.150	0.171	0.195	0.239	0.284	0.328
	30,000	0.166	0.182	0.208	0.246	0.278	0.342	0.406	0.470
500 deg	2,000	0.001	0.001	0.001	0.001	0.002	0.002	0.002	0.003
	3,000	0.002	0.002	0.003	0.004	0.004	0.004	0.005	0.006
	4,000	0.004	0.004	0.005	0.006	0.006	0.008	0.009	0.011
	6,000	0.008	0.009	0.011	0.012	0.014	0.017	0.020	0.023
	8,000	0.015	0.016	0.019	0.022	0.025	0.031	0.036	0.042
	10,000	0.023	0.025	0.030	0.034	0.039	0.047	0.056	0.065
	15,000	0.052	0.057	0.067	0.077	0.087	0.107	0.127	0.146
	20,000	0.092	0.101	0.119	0.137	0.155	0.190	0.226	0.260
	25,000	0.145	0.158	0.186	0.213	0.242	0.297	0.354	0.408
	30,000	0.215	0.235	0.277	0.317	0.360	0.443	0.526	0.610
750 deg	2,000	0.001	0.001	0.001	0.002	0.002	0.002	0.003	0.003
	3,000	0.003	0.003	0.003	0.004	0.004	0.005	0.006	0.007
	4,000	0.005	0.005	0.006	0.007	0.008	0.009	0.011	0.013
	6,000	0.010	0.011	0.013	0.015	0.017	0.021	0.025	0.029
	8,000	0.018	0.020	0.024	0.027	0.030	0.038	0.045	0.052
	10,000	0.029	0.031	0.037	0.042	0.048	0.059	0.070	0.081
	15,000	0.064	0.070	0.083	0.095	0.118	0.133	0.158	0.182
	20,000	0.114	0.125	0.148	0.170	0.192	0.236	0.281	0.324
	25,000	0.180	0.196	0.231	0.265	0.300	0.370	0.440	0.508
	30,000	0.256	0.281	0.322	0.381	0.430	0.530	0.628	0.730
1000 deg	2,000	0.001	0.002	0.002	0.002	0.002	0.003	0.003	0.004
	3,000	0.003	0.003	0.004	0.005	0.005	0.006	0.008	0.009
	4,000	0.006	0.006	0.007	0.008	0.009	0.011	0.014	0.016
	6,000	0.012	0.014	0.016	0.018	0.021	0.025	0.030	0.035
	8,000	0.022	0.024	0.028	0.033	0.037	0.045	0.054	0.062
	10,000	0.034	0.038	0.044	0.051	0.057	0.071	0.084	0.097
	15,000	0.077	0.084	0.099	0.114	0.129	0.159	0.189	0.212
	20,000	0.127	0.150	0.177	0.204	0.230	0.283	0.336	0.390
	25,000	0.215	0.235	0.277	0.317	0.360	0.443	0.526	0.610
	30,000	0.307	0.337	0.386	0.456	0.515	0.634	0.753	0.870

Table 4—Total Draft Losses for Design of Individual
at 4 Percent CO_2 in the Vent

Four Percent CO_2		L/D; Ft. of Con			
Fig. 2	Vertical Only	5	10	20	30
Fig. 1	Connector Used	—	—	—	—
Average Temperature Rise Above Ambient of Gases in Vertical Part of Stack or Chimney, F Degrees Above 60 F Ambient	Input Loading: Btu Per Hour Per Square Inch of Vent or Chimney Area: I/A				
100 deg	4,000	0.008	0.009	0.010	0.012
	6,000	0.018	0.020	0.024	0.026
	8,000	0.032	0.035	0.042	0.048
	10,000	0.050	0.055	0.065	0.075
	15,000	0.114	0.122	0.147	0.169
	20,000	0.202	0.222	0.262	0.300
	25,000	0.317	0.348	0.410	0.471
	30,000	0.454	0.498	0.586	0.675
	40,000	0.810	0.890	1.05	1.20
200 deg	4,000	0.009	0.010	0.012	0.014
	6,000	0.021	0.023	0.027	0.031
	8,000	0.038	0.041	0.049	0.056
	10,000	0.059	0.064	0.076	0.087
	15,000	0.132	0.145	0.170	0.196
	20,000	0.235	0.256	0.303	0.349
	25,000	0.368	0.404	0.475	0.547
	30,000	0.526	0.578	0.680	0.782
	40,000	0.940	1.03	1.21	1.40
300 deg	4,000	0.011	0.012	0.014	0.016
	6,000	0.024	0.026	0.031	0.036
	8,000	0.043	0.047	0.056	0.064
	10,000	0.067	0.073	0.086	0.099
	15,000	0.150	0.165	0.194	0.223
	20,000	0.268	0.294	0.346	0.400
	25,000	0.420	0.460	0.541	0.623
	30,000	0.600	0.660	0.775	0.89
	40,000	1.07	1.18	1.38	1.59
400 deg	4,000	0.012	0.013	0.016	0.018
	6,000	0.027	0.030	0.035	0.040
	8,000	0.048	0.053	0.063	0.072
	10,000	0.075	0.083	0.097	0.112
	15,000	0.169	0.186	0.219	0.252
	20,000	0.301	0.331	0.389	0.448
	25,000	0.472	0.518	0.610	0.701
	30,000	0.675	0.741	0.871	1.00
	40,000	1.21	1.32	1.56	1.79
500 deg	4,000	0.013	0.015	0.017	0.020
	6,000	0.030	0.033	0.039	0.044
	8,000	0.053	0.059	0.069	0.079
	10,000	0.083	0.091	0.107	0.124
	15,000	0.187	0.205	0.241	0.278
	20,000	0.333	0.365	0.429	0.495
	25,000	0.52	0.572	0.672	0.795
	30,000	0.75	0.82	0.96	1.11
	40,000	1.33	1.46	1.72	1.98

ᵃ Not recommended for design of systems smaller than 8 in. inside diameter.

GAS VENT SYSTEMS HAVING DRAFT HOODS TO OPERATE
GASES[a], INCHES OF WATER COLUMN

SYSTEM LENGTH TO DIAM. RATIO
NECTOR AND STACK PER FOOT OF DIAMETER

40	60	80	100	120	140	160
10	30	50	70	90	110	130
0.014	0.017	0.020	0.023	0.026	0.030	0.033
0.031	0.038	0.045	0.052	0.059	0.066	0.073
0.054	0.067	0.080	0.092	0.105	0.118	0.131
0.085	0.104	0.124	0.144	0.163	0.183	0.204
0.187	0.235	0.279	0.323	0.368	0.411	0.460
0.340	0.418	0.497	0.575	0.652	0.731	0.816
0.531	0.655	0.780	0.901	1.03	1.15	1.28
0.760	0.936	1.11	1.29	1.46	1.64	1.83
1.36	1.68	1.99	2.30	2.62	2.93	3.26
0.016	0.019	0.023	0.027	0.030	0.034	0.038
0.035	0.044	0.052	0.060	0.068	0.076	0.085
0.063	0.078	0.092	0.107	0.122	0.136	0.151
0.099	0.121	0.144	0.167	0.190	0.212	0.235
0.222	0.273	0.324	0.375	0.426	0.476	0.529
0.395	0.486	0.576	0.669	0.760	0.850	0.940
0.620	0.761	0.902	1.05	1.19	1.33	1.47
0.885	1.09	1.29	1.50	1.70	1.90	2.11
1.58	1.94	2.31	2.67	3.04	3.40	3.76
0.018	0.022	0.026	0.030	0.035	0.039	0.043
0.040	0.050	0.059	0.069	0.078	0.087	0.096
0.072	0.089	0.105	0.122	0.139	0.155	0.172
0.112	0.138	0.164	0.190	0.216	0.242	0.268
0.253	0.311	0.369	0.429	0.485	0.554	0.603
0.450	0.554	0.657	0.761	0.865	0.968	1.07
0.705	0.869	1.03	1.20	1.36	1.52	1.68
1.01	1.24	1.47	1.71	1.94	2.17	2.4
1.80	2.21	2.63	3.05	3.46	3.87	4.30
0.020	0.025	0.030	0.034	0.039	0.044	0.048
0.045	0.056	0.066	0.077	0.087	0.098	0.108
0.081	0.100	0.118	0.137	0.156	0.175	0.193
0.126	0.155	0.184	0.213	0.242	0.272	0.300
0.284	0.350	0.415	0.48	0.546	0.611	0.678
0.505	0.622	0.740	0.855	0.971	1.09	1.21
0.792	0.975	1.16	1.34	1.52	1.71	1.89
1.13	1.40	1.66	1.92	2.18	2.44	2.70
2.02	2.49	2.96	3.42	3.89	4.36	4.82
0.022	0.028	0.033	0.038	0.043	0.048	0.053
0.050	0.062	0.073	0.085	0.096	0.108	0.120
0.089	0.110	0.131	0.152	0.172	0.193	0.214
0.139	0.171	0.203	0.236	0.268	0.300	0.333
0.314	0.386	0.458	0.531	0.604	0.675	0.750
0.558	0.688	0.815	0.946	1.07	1.20	1.33
0.875	1.08	1.28	1.48	1.68	1.89	2.09
1.25	1.54	1.83	2.12	2.41	2.70	2.99
2.23	2.75	3.26	3.78	4.30	4.81	5.33

1. *Friction* is represented by the ratio L/D where L is the total piping length of connector plus stack, in feet; D is the diameter of the connector and stack, in feet (both are assumed to have the same diameter).

The greater the system length for a given diameter, the greater will be the energy losses. Values in these tables for systems having a connector include the effect of the energy losses in a tee, elbow or breeching. For vertical vents (Fig. 2) there is, of course,

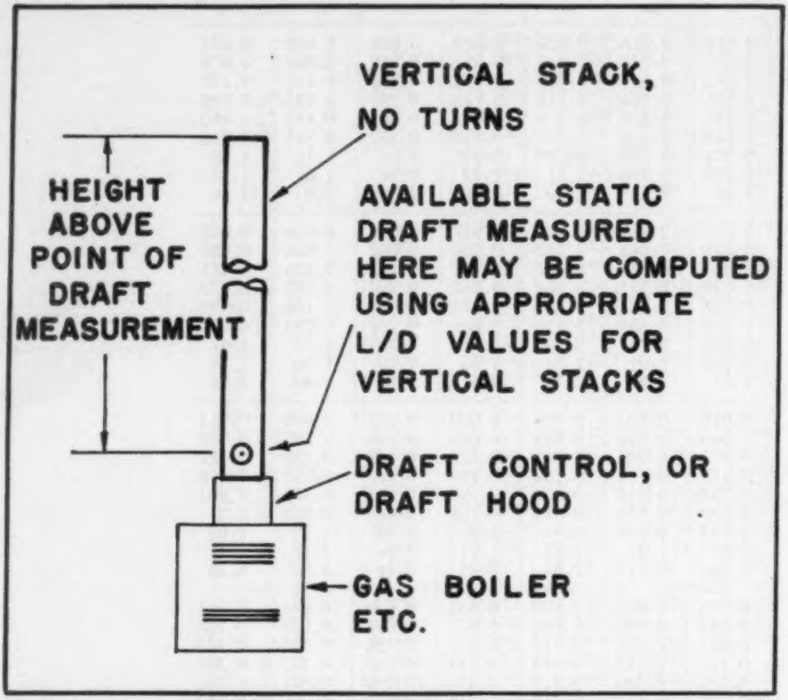


FIG. 2—DRAFT MEASUREMENT IN VERTICAL STACK HAVING NO HORIZONTAL BREECHING

no connector or tee and $L = H$, or the length of piping is equal to stack height. Where connector and stack or chimney sizes differ, use the smaller area to find D, and L/D.

2. *Input and cross sectional area* is represented by the ratio I/A where I is the hourly fuel input or rate of gas consumption, Btu per hour; A is the area of connector stack, chimney or vent, square inches.

The ratio I/A could be called the input loading or Btu input per square inch of flue area. This factor is frequently used as a basis for gas chimney design and serves to relate heat input to flow velocity. Table 5 converts area to diameter, to assist in the calculations. Where connector area differs from stack or chimney area, the smaller area should be used to determine both the I/A and L/D ratios.

TABLE 5—CONVERSION FROM SQUARE INCHES AREA TO ROUND DIAMETER IN FEET

AREA SQ. IN., A[a]	DIAM. FEET, D	AREA, SQ. IN., A[a]	DIAM. FEET, D
50	0.67	4070	6
113	1	5540	7
254	1.5	7230	8
453	2	9250	9
707	2.5	11300	10
1020	3	25400	15
1800	4	45300	20
2820	5		

[a] $A = 113\ D^2$.

3. *Average gas temperature rise above ambient* in the vertical stack. This is represented in Tables 2 and 3 by the temperature rises of 150, 300, 500, 750, and 1000 deg in the blocks in the left hand margin. Table 4 for gas vents goes from 100 to 500 deg above room. Energy losses, as shown by the draft equation, increase with temperature due to greater flow velocity. Precise work also requires that average temperature be compensated for cooling in accordance with Table 6. For chimney and vent design, ambient temperature should be chosen as 60 F outdoors. Temperature in the boiler room does not influence chimney operation. If the chimney design is adequate for 60 F operation, any drop in outdoor temperature will improve stack action because of greater difference between stack and ambient temperature. Choice of a lower chimney operating temperature than that actually expected will yield a conservative design, with extra capacity for overloads.

4. *Gas composition or CO_2 concentration* (on a dry basis). The velocity of stack gas flow for a given heat input increases as the CO_2 in the gases decreases, thereby increasing the energy losses at higher amounts of excess air. Table 2 for 4 percent CO_2 and Table 3 for 8 percent represent 2 common operating conditions of stacks required to produce draft. For other operating conditions, at other values of CO_2, Table 7 can be used to

TABLE 6—EFFECT OF CHIMNEY GAS TEMPERATURE UPON RELATIVE DRAFT LOSSES (BASED ON 300 DEG GAS TEMPERATURE RISE ABOVE 60 F AMBIENT)

AVG TEMP RISE OF CHIMNEY GASES, F DEG	RELATIVE DRAFT LOSS	AVG TEMP RISE OF CHIMNEY GASES, F DEG	RELATIVE DRAFT LOSS
50	0.70	700	1.49
100	0.76	800	1.61
200	0.88	900	1.73
300	1.00	1000	1.85
400	1.12	1200	2.10
500	1.24	1500	2.46
600	1.37	2000	3.08

Effect of cooling of gases in connector, breeching and vertical section: Estimate cooling as

1. Steel connector and stack	4 % per 10 diam
2. Masonry chimney	1 % per 10 diam

correct the losses in either Table 2 or 4 for 4 percent CO_2. Thus with 12 percent CO_2 obtained during theoretical combustion of natural gas with no excess air, Table 7 shows the draft loss to be 0.137 times that given for 4 percent CO_2, or roughly $^1/_7$ as great due to lowered gas velocity in the system.

Two examples of stack design calculations will be given, *viz.* (1) Gas boiler requiring a definite static draft; (2) Gas applicance having a draft hood and thus requiring zero static draft at the inlet of the draft hood, or at the appliance flue outlet.

The following information should be obtained and noted about the equipment for which a stack or chimney is required:

1. Heat input, Btu per hour, of fuel gas consumed.
2. Temperature of gases delivered to connector at equipment outlet.
3. CO_2 concentration of gases in chimney or vertical stack. This can depend on whether the draft control acts to bleed ambient air into the stack, or operates an internal movable damper solely to increase resistance, as well as on equipment operational characteristics.

TABLE 7—EFFECT OF CO_2 CONCENTRATION UPON RELATIVE DRAFT LOSSES (USE WITH TABLE 2 FOR 4 PERCENT CO_2)

CO_2 PERCENT IN CHIMNEY GASES	RELATIVE DRAFT LOSS	CO_2 PERCENT IN CHIMNEY GASES	RELATIVE DRAFT LOSS
1	14.8	7	0.352
2	3.8	8	0.279
3	1.75	9	0.224
4	1.00	10	0.186
5	0.655	11	0.157
6	0.455	12	0.137

4. Static draft required at the boiler or furnace outlet, or whether there is a draft hood.

For stacks or gas vents installed directly on top of the draft control or draft hood as shown in Fig. 2, available draft, or flow, must be computed using L/D values for vertical vents. For systems having lateral connectors care must be taken where Fig. 1 is used, to choose the lower L/D line in Tables 2, 3, and 4 in order to obtain proper compensation for connector and fitting resistance losses.

If the connector or breeching has 2 elbows as in Fig. 3, the point of draft measurement is as indicated thereon. Available draft at the draft control may be greater or less than that calculated, depending on configuration of the system. The Appendix includes a point-by-point analysis of this specific type of system, when operating as a gas vent, which shows that available static draft at the draft hood outlet is actually less than at the base of the final vertical section.

Very frequently a boiler or equipment manufacturer will specify a breeching or connector size which differs from that of the vertical stack. It may be either larger or smaller. The same situation may be encountered when a replacement boiler is connected to an existing chimney. Where there are 2 different piping areas in such instances, simply compute the draft or design the system as previously stated on the basis of:

1. Total lineal feet of piping—connector and stack; and
2. Smaller area, regardless of whether it is the connector, or the vertical stack.

Theoretical exploration of the resistances of connectors, stacks, and breechings, having different sizes indicates that a controlling influence on draft or flow is exerted by losses in that part of the system having the smaller area. If the system losses are figured for the smaller area, actual losses will be slightly less. Thus use

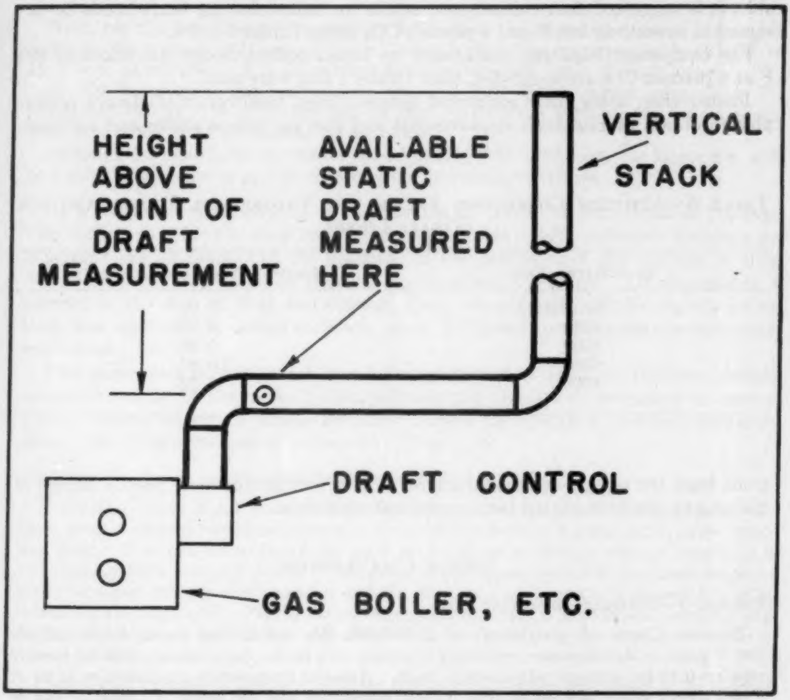


FIG. 3—DRAFT MEASUREMENT IN A SYSTEM HAVING TWO 90 DEG TURNS

of the smaller area for design or draft calculation alwa·s yields a conservative design. This is true whether there is a great disparity in sizes, or where they are nearly the same.

L = total lineal feet of piping.
D = smaller diameter in system (or equivalent hydraulic diameter for shapes other than round. Hydraulic diameter = 4 × cross section area/ Length of wetted perimeter.

Similarly, the input loading I/A must be found using the smaller area.

In making these calculations, it is a good idea to allow adequately for changes in

such factors as outdoor air temperature, heat input and excess air, by designing for more draft than the minimum required. It is always possible to find some means to reduce draft. But a chimney or stack which is too short will never be able to produce more than a certain amount of draft and one that is undersized will rapidly build up excessive energy losses. Conversely, it is uneconomical to build a stack of excessive height or diameter, which is both costly as well as wasteful of heat.

For a choice of design operation conditions where no other information is available, it is suggested that chimneys or stacks for boilers having draft hoods be designed to operate at 300 F and 4 percent CO_2 using Tables 1 and 4.

For equipment requiring draft, such as steam boilers, design conditions of 500 F at 8 percent CO_2 are suggested, thus Tables 1 and 3 are used.

Rather than using these estimated temperatures, however, it is always preferable to obtain precise draft requirements and flue gas temperatures and compositions from the equipment manufacturer. This will permit more precise design of the proper stack to obtain best equipment operation.

TABLE 8—ALTITUDE CORRECTION FACTOR FOR THEORETICAL DRAFT, AND FOR DRAFT LOSSES

At Altitude, Feet	Multiply Draft and Losses By
Sea Level	1.00
2000	0.93
4000	0.86
6000	0.81
8000	0.75

SAMPLE CALCULATIONS

Example I—Designing to produce draft.

Problem: Given—A gas furnace of 22,000,000 *Btu per hr heat input, which delivers* 380 *F gases to the connector containing* 8 *percent* CO_2 *in the flue products, and the furnace requires* 0.10 *in. of water column static draft. Ambient temperature is assumed to be 60 F. This furnace has a side outlet (Fig.* 1) *vent connection and the length of connector will be* 20 *ft.*

Find: (*a*) *The height of steel stack required and* (*b*) *The diameter of stack required.*

Solution: Refer to Table 1. Assume an average gas temperature rise in the vertical stack of 300 deg, allowing 20 deg cooling from an inlet rise of 320 deg as a first guess, the lowest height to produce 0.10 in. theoretical draft will be 20 ft, at which 0.11 in. H_2O is obtained. This allows only 0.01 in. for losses, which is too slim a margin for operating variations. Thus, try 25 ft of height at which the theoretical draft is 0.13 in.

Proceed to find the losses: With 20 ft of connector and 25 ft of stack height, $L = 45$ ft. Next, refer to Table 3, 8 percent CO_2. Estimate that L/D will be about 30, using the L/D line for stacks having connectors (always try this value of L/D for the first guess, since it is in about the middle of the Table). In the group of numbers for 300 deg gas temperature rise read down to the value of I/A where the losses are just less than the

theoretical draft minus the draft needed. In this case, the losses must not exceed 0.03 in. H_2O. Using Table 3, at

$$I/A = 10{,}000,\ \text{losses} = 0.027 \text{ in. } H_2O$$

To find stack area, divide input by I/A at which the losses are 0.03 in., thus

$$\frac{22{,}000{,}000}{10{,}000} = 2{,}200 \text{ sq in., which is the approximate stack area needed.}$$

Refer to the auxiliary Table 5. This shows that a 5-ft diameter stack of 2,820 sq in. area might be pretty close.

Next, use the 5-ft diameter to recheck L/D, since the approximate area just found indicates the original L/D assumption to be too high. The corrected value of $L/D = 45/5 = 9$, so use the $L/D = 10$ column to estimate a new value of draft losses.

At 300 deg average gas temperature rise, and $L/D = 10$, (again using Table 3), losses = 0.020 at $I/A = 10{,}000$ leaving $0.13 - 0.020 = 0.11$ in. theoretical draft. This shows that the 5-ft diameter is adequate.

Although a stack diameter smaller than 5 ft might be used here, the larger size will be a safer design, due to greater tolerance for operating variations.

As a matter of final interest, it is desirable to check for the effects of cooling. The cooling rate for the steel stack is given in Table 6 with sufficient accuracy as 4 percent per 10 diam, but the distance to the midpoint of the vertical is only 32.5/5 or about 6½ diam, so that cooling is about 3 percent. 320 deg minus 3 percent is 310 deg, so that the average stack temperature will be slightly more than that used for the design estimate, again providing some margin for operating variations.

This same sort of calculation can be applied to any stack or chimney design within the scope of the tables. Interpolation can be used as necessary or curves plotted where needed to permit chimney designs for operation at other temperatures, CO_2 concentrations, or values of I/A or L/D.

Example II—Design for zero draft (where a draft hood is used).

Problem:—Given: A gas boiler of 6,000,000 *Btu per hr input, having a draft hood, delivers gases at* 480 *deg rise above ambient to the draft hood inlet, at* 8 *percent* CO_2 *in the boiler flue gases. It is desired to design the stack to operate at a dilution ratio of roughly* 2.0 (2 *pounds of stack gases per pound of flue gases) meaning that there will be a stack temperature of not over* 240 *deg above ambient and the stack gases will contain* 4 *percent* CO_2 *after dilution at the draft hood. Piping arrangement must consist of a* 10*-ft lateral connector and* 80 *ft of height above the connector breeching in order to discharge the stack gases above a nearby building.*

Find: *Stack diameter to produce sufficient flow to properly vent this boiler at the desired dilution ratio.*

Solution: Because the draft hood is used, this system must be designed using Tables 1 and 4 to obtain losses exactly equal to the theoretical draft.

Assume an average temperature of 200 F in the vertical stack, (this allows 40 deg cooling as a first guess) Table 1 shows that theoretical draft will be 0.33 in. at 80 ft. of height.

The diameter and thus the L/D ratio for this system is not known. As a first guess, try $L/D = 30$ using the L/D line for stacks with connectors to find what value of I/A will produce draft losses of 0.33 in. H_2O.

In Table 4, the column for $L/D = 30$ for stacks with connectors shows the following losses:

$$I/A = 15{,}000,\ \text{losses} = 0.273 \text{ in. } H_2O$$
$$I/A = 20{,}000,\ \text{losses} = 0.486 \text{ in. } H_2O$$

Interpolating to find the exact value of I/A at which losses = 0.33 in.

$I/A = 15{,}000 + (0.33 - 0.273)/(0.486 - 0.273) \times 5{,}000 = 16{,}340$ Btu per sq in.

Solving for the stack area, divide input by I/A,

6,000,000 Btu/16,340 Btu/sq in. = 367 sq in. or roughly the area of a 22-in. diam stack.

Making a second approximation, on the basis that the 22-in. computed diameter permits finding a corrected L/D,

$$\text{New } L/D = 90 \times 12/22 = 49 \text{ or } 50 \text{ for practical purposes}$$

Again at 200 deg, but in the column for $L/D = 50$ in a stack with a connector, losses = 0.324 in. H_2O at $I/A = 15{,}000$. For all practical purposes this is equal to the 0.33 in. of theoretical draft, so the final value of I/A is 15,000 Btu of appliance input per sq in. of stack area.

Computing stack area:

$$A = 6{,}000{,}000/15{,}000 = 400 \text{ sq in.}$$
$$D = \sqrt{400/0.785} = 22.6 \text{ in.}$$

For this stack, a diameter of 23 in. may be used. Because the second approximation changed D by only 0.6 in., it may readily be seen that neither L/D nor diameter would change appreciably upon a third approximation.

Checking cooling for the 23-in. stack to determine whether the midpoint temperature will be below the assumed value, the length of stack to the midpoint is $(10 + 40)12/23$ diam, or 26 diam. The cooling rate is 4 percent per 10 diam. This will produce roughly 11 percent cooling for the steel stack. Eleven percent of 240 is 27 deg, so that $240 - 27$ is 213 deg. This means that the midpoint temperature will be greater than the design temperature, thus assuring adequate stack gas flow.

For vertical stacks, the upper L/D line in Table 4 must be used to find the diameter. Elimination of fitting losses means that a given diameter of straight vertical stack is capable of carrying more flow, or conversely for any value of static draft loss, the permissible L/D is much greater. The difference in the L/D headings for stacks with and without connectors shows that elimination of the tee makes a difference of one velocity head or 30 L/D.

In case there is any doubt as to whether the stack will be vertical or have fittings, it should be designed as a system with fittings and having a connector. Should some or all of the fittings be eliminated, the added flow will not impair operation.

The 2 examples just discussed, illustrate how the draft equation may be solved quickly by cut-and-try methods and simple interpolation. A few trials will permit rapid estimation of stack, chimney, or vent sizes with very little effort, as the user becomes familiar with the process of making educated guesses. The method of solution results in convergence so that even if the first guess for L/D is way off, using the results in the second trial will yield a reasonably close answer.

CONCLUSION

A simple, general method of tabulating parameters of gas equipment chimney operation has been developed which permits rapid design of vents, stacks and chimneys for all types of gas equipment. The method applies irrespective of draft or chimney efficiency requirements. Thus vents for gas appliances having a draft hood may be designed as readily as high large diameter chimneys for large industrial gas furnaces or boilers, requiring strong stack draft.

REFERENCES

1. HEATING, VENTILATING, AIR CONDITIONING GUIDE 1956, Chapter 17—Chimneys and Draft Calculations, Equation 3, p. 424 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.).

2. Performance of Residential Chimneys, by L. B. Schmitt and R. B. Engdahl (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 241).
3. Operating Characteristics of a Gas Vent, by A. Kinkead (*Pacific Coast Gas Association Proceedings*, Vol. 43, 1952).

APPENDIX

DERIVATION OF THE CHIMNEY DRAFT EQUATION

The chimney draft equation may be derived by use of applicable portions of Bernoulli's equation, as was done in reference 3. For present purposes, it appears desirable to consider the change in total head comprising the sum of kinetic, potential and pressure energy.

At any level within a chimney system the total flow energy or head, h, may be represented by;

$$h = \frac{V^2}{2g} + H\left(\frac{T_2}{T_1} - 1\right) + \frac{P}{\rho} \quad \text{(A-1)}$$

where
h = total head, feet of fluid flowing.
V = flow velocity, feet per second.
T_2 = average temperature of gases in vertical portion of vent or chimney, Fahrenheit, absolute.
T_1 = temperature of ambient air, Fahrenheit, absolute.
P = static pressure in chimney, pounds per square foot.
ρ = density of chimney gases at T_2, pounds per cubic foot.
H = distance above the datum, feet.

Consider now a chimney of constant diameter in which the hot gases lose no heat to the atmosphere (so that T_2 is constant). The chimney gases flow vertically from station 1, at which the draft is being determined, up to station 2 which is just below the chimney outlet. No top or cowl is used on the chimney, so that the gases are discharged vertically to atmosphere.

Because the chimney gases are lighter than air, there will be a buoyant head at station 1, but none at station 2. Thus it is desirable to choose the datum for determining buoyant head at the level of the outlet. This reverses signs, or means that H_1 is actually measured down from the outlet.

At draft station 1, at the bottom of the chimney, the sum of the terms comprising total head h_1 is:

$$h_1 = \frac{V_1^2}{2g} + H_1\left(\frac{T_2}{T_1} - 1\right) + \frac{P_1}{\rho} \quad \text{(A-2)}$$

At outlet station 2, the sum of the terms comprising total head h_2 is

$$h_2 = \frac{V_2^2}{2g} + H_2\left(\frac{T_2}{T_1} - 1\right) + \frac{P_2}{\rho} \quad \text{(A-3)}$$

Frictional energy losses in the chimney channel between stations 1 and 2 result in a reduction in total head, thus to equate h_1 and h_2 one can write

$$h_1 = h_2 + LH \quad \text{(A-4)}$$

where

LH = lost head

Combining Equations A-2, A-3 and A-4 the equation for the head balance between stations 1 and 2 is obtained, thus:

$$\frac{V_1^2}{2g} + H_1\left(\frac{T_2}{T_1} - 1\right) + \frac{P_1}{\rho} = \frac{V_2^2}{2g} + H_2\left(\frac{T_2}{T_1} - 1\right) + \frac{P_2}{\rho} + LH \quad . \quad \text{(A-5)}$$

This equation may be simplified by the following assumptions.

(a) With the chimney gases flowing in a channel of constant area, and at constant temperature: $V_1 = V_2$, $V_1^2/2g = V_2^2/2g$ and these terms cancel out.

(b) H_2 is zero since the station 2 is defined as being just below the chimney outlet, and the difference may be expressed as H, which is chimney height above point of draft measurement. $H_1 - H_2 = H$.

(c) Lost head will be expressed as LH $= R_e\, V^2/2g$ where R_e is resistance of the flow channel between stations 1 and 2, in velocity heads.

Rewriting Equation A-5 to include the simplification

$$\left(\frac{P_1}{\rho} - \frac{P_2}{\rho}\right) + (H_1 - H_2)\left(\frac{T_2}{T_1} - 1\right) = \text{LH} \quad . \; . \; . \; . \; . \quad \text{(A-6)}$$

$$\frac{P_1 - P_2}{\rho} + H\left(\frac{T_2}{T_1} - 1\right) = R_e\frac{V^2}{2g} \quad . \; . \; . \; . \; . \; . \quad \text{(A-7)}$$

In Equation A-7, the pressure difference $(P_1 - P_2)$ is available static draft, providing that the buoyant force exceeds resistance losses, so that absolute pressure in the chimney at station 1 is below atmospheric. For convenience then it is possible to write,

$$\frac{P_2 - P_1}{\rho} = H\left(\frac{T_2}{T_1} - 1\right) - R_e\frac{V^2}{2g} \quad . \; . \; . \; . \; . \; . \quad \text{(A-8)}$$

so that if the chimney produces draft, the draft will be in the positive sense if the buoyant term exceeds energy losses from point of draft measurement to the system outlet.

Converting from $(P_2 - P_1)$ in lb per sq ft, to draft in inches of water column, S:

$$S = \frac{0.255P}{T_2}\left[H\left(\frac{T_2}{T_1} - 1\right) - R_e\frac{V^2}{2g}\right] \quad . \; . \; . \; . \; . \; . \quad \text{(A-9)}$$

where

S = available static draft, inches of water column.
P = barometric pressure, inches of mercury.
T_2 = average temperature of gases in vertical portion of vent or chimney, Fahrenheit, absolute.
H = height of chimney outlet above point of draft measurement, feet.
T_1 = Ambient temperature of outdoor air, Fahrenheit, absolute.
R_e = resistance of connector and chimney system, between point of draft measurement, and outlet of flue, velocity heads.
V = average velocity of chimney gases, at temperature T_2, feet per second.
g = 32.2 ft per second per second, acceleration of gravity.

In Equation A-9, the temperature at the midpoint of the vertical section may be taken as T_2, making allowances for heat losses on the way.

Connector and chimney resistance for use in this equation when designing for draft, as expressed in velocity heads, comprises friction and fitting losses, but does not include the customary one velocity head for acceleration, since this head cannot be included or measured as available static draft.

The value of R_e may be found from:

$$R_e = F + \frac{4fL}{D} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{(A-10)}$$

where

F = velocity heads of fitting losses.
f = Fanning friction factor for the piping.
L = length of all piping in system, from point of draft measurement to top of stack or chimney, feet.
D = inside diameter of connector or chimney piping, whichever is the smaller, feet.

Assuming a value of $f = 0.0075$ Equation A-10 becomes, for stacks and chimneys

$$R_s = F + 0.03\frac{L}{D} \qquad \text{(A-10a)}$$

The value of f is dependent on the Reynolds number of the gases in the stack, which is diameter times gas velocity times gas density, divided by gas viscosity, all in consistent units, and on roughness of the interior surface. Choice of a suitable compromise value for the friction factor f is rendered difficult by the wide span of variation of the Reynolds number in stacks and chimneys and gas vents. Even with a Reynolds number variation from 4000 to 100,000, however, the value of f changes only from 0.01 to 0.005 for smooth surfaces. For greater relative surface roughness, the value of f also increases above those values at any Reynolds number.

Studies of materials commonly used for gas vents indicates that in small sizes from 3-in. to 6-in. diameters, the friction factor probably does not exceed 0.0083 at Reynolds numbers from 8000 to 15,000. For larger sizes of connector and chimney piping, and at economic gas flow velocities the friction factor of 0.0075 used for the preceding tables will either be very close, or conservative.

It cannot be said that the friction factor of 0.0075 applies to the usual rough masonry work encountered in domestic chimney construction, where offset bricks, extruded mortar, out of line tile and other probable constructional faults reduce the effective area as well as impede flow. It is also not usually necessary to choose a friction factor which allows for future soot accumulation, because gas-fired equipment is not normally subject to this difficulty. The responsible chimney designer should therefore make certain that the constructional practices employed for chimneys result in smooth interior surfaces so that design values of flow and draft can be attained.

In addition to fittings such as tees or elbows between the point of draft measurement and the top of the chimney system, there is usually a length of connector piping between the equipment and the breeching, tee, or elbow inlet to the vertical portion. To include the length of vent connector in the total resistance (assuming that it is the same size as the vertical vent) the length L includes the length of all piping in the system, measured along the piping centerline, between the point of measurement and outlet.

Resistance values of common fittings in velocity heads may be found in many references on duct and piping system design. Elbows for example may vary from 0.5 to 1.4 velocity heads, while tee fittings and related breechings can range from 1.0 to 2.0 For Table 4 previously given for gas vents, resistances used were as shown in Table A-1.

For vent or chimney systems in which the fitting resistance is greater or less than the values chosen for these tables, there will be a different scale of L/D versus resistance. The increments of L/D values in the headings, however, will be unchanged. For example, a gas vent may have 3.2 velocity heads of total resistance, but the L/D value may be only 20, as a result of a resistance equation in which fixed and acceleration resistance totals 2.6 instead of 2.4 velocity heads.

Precision of this order is seldom required in vent or chimney design, because adequate capacity or draft to compensate for unknown resistances can always be achieved by the simple expedient of increasing the size.

It must be assumed that adequate provision has been made for air supply into the boiler or furnace room for combustion, draft control dilution and general ventilation. Various rules exist for sizing of air supply openings, however the simplest treatment assumes that the opening is an orifice in series with the vent system. The resistance of

an orifice is 2.6 velocity heads relative to its own area. Thus an orifice of the same size as the vent would also have a resistance of about 2.6 velocity heads.

An air supply of any area, A_a, will have a resistance of 2.6 $(A_v/A_a)^2$ where A_v is vent area. If the air supply is reduced much below vent area, chimney draft may be seriously impaired. For example, if A_v/A_a is 2.0, air supply resistance becomes 10.4 velocity heads. Usually the air supply through any special opening is augmented by general infiltration, the amount of which depends on the building size and type of construction. Thus the true or effective air supply resistance affecting vent operation can be computed only if the total area, including infiltration crackage, is known.

Having covered the factors involved in choosing proper resistance values, it is now necessary to convert velocity, V, into more convenient terms. Velocity of gas flow in a connector, stack or chimney may be expressed as a function of fuel input, CO_2 piping

TABLE A-1—RESISTANCES USED IN TABLE 4

Acceleration head	1.0	Velocity Heads
Entrance (draft hood) loss	0.4	" "
Piping losses	$0.03\ L/D$	" "
Fitting loss (elbow or tee)	1.0	" "
Vertical gas vents	$R_v = 1.4 + 0.03\ L/D$, total	
Gas vents with connectors	$R_v = 2.4 + 0.03\ L/D$, "	

diameter, and temperature. The weight of flue gas produced per hour by burning natural gas, or other gases of similar carbon to hydrogen ratios, at a heat input of I Btu per hr is roughly

$$M = \frac{7.2\ I}{10000}\left(0.15 + \frac{11}{CO_2}\right)^{*} \quad . . . . \quad \text{(A-11)}$$

where

M = flue gases produced per hour, pounds.
CO_2 = percent CO_2 that these gases contain.
I = Heat input, gross, Btu per hour.

Assuming a volume of 13.1 cu ft per lb for flue gases at standard conditions of 60 F and 30 in. of mercury (not compensated for change in specific gravity with CO_2), actual velocity in feet per second, V at T_2, in a flue of an area of A square feet, becomes:

$$V = \left[\frac{2.62\ I\ (0.15 + (11/CO_2))}{A \times 10^6}\right]\frac{T_2}{T_1} \quad . . . . \quad \text{(A-12)}$$

Finally, Equations A-10a and A-12 may be substituted into Equation A-9 to obtain a single expression for available static draft:

$$S = \frac{0.255\ P}{T_2}\left\{H\left(\frac{T_2}{T_1} - 1\right) - \frac{(F + 0.03\ (L/D))}{64.4}\left[\frac{2.62\ I\ (0.15 + (11/CO_2))}{A \times 10^6}\frac{T_2}{T_1}\right]^2\right\} \quad . . . \quad \text{(A-13)}$$

* For natural gases of 950 to 1100 Btu per cu ft., see Fundamentals of Heat Transfer in Domestic Gas Furnaces, by R. L. Stone, *Amer. Gas Assn. Laboratories Research Bulletin*, No. 63, May 1951, Equation. 87, p. 129.

The following example indicates how this equation may be applied directly to the computation of available static draft.

Example of Draft Computation: At sea level (P = 30-in. Hg), a 5,000,000 Btu per hr input furnace (I = 5,000,000) having a 2-ft diameter flue (D = 2) and connector will operate at 300 deg F (T_2 = 820 F, absolute) above ambient temperature of 60 F (T_1 = 520 F, absolute) at 8 percent CO_2 in the flue products of natural gas. Its stack height will be 100 ft (H = 100) and its connector will be 25 ft long (L = 125) (Use Fig. 1), considering the breeching as a tee fitting (F = 1.4 velocity heads) what will the draft be? The following supplementary data is also available: area of flue, (A = 3.14 sq ft); resistance, (R_s = 1.4 + 0.03 × 125 = 3.27 velocity heads), between point of draft measurement and the stack outlet. Substituting in Equation A-13:

$$S = \frac{0.255 \times 30}{820}\left[100\left(\frac{820}{520} - 1\right) - \frac{3.27}{64.4}\left(\frac{2.62 \times 5 \times 10^6 \times (0.15 + 11/8)}{3.14 \times 10^6}\,\frac{820}{520}\right)^2\right]$$

S = 0.491 inches water column available static draft at appliance.

FACTORS IN TABLES 1 THROUGH 7

Factors used in Equation A-13 to compute the tables were as follows:

Table 1: Theoretical static draft $= \dfrac{0.255\,P}{T_2}\,H\left(\dfrac{T_2}{T_1} - 1\right)$, in. H_2O . . (A-13a)

P = 30 in. Hg.
T_1 = 520 F, absolute.
H and T_2 = as given in Table.

Tables 2 and 3: Static draft pressure losses, in inches H_2O.

$$\text{Losses} = \frac{0.255\,P}{T_2}\left(\frac{F + 0.03L/D}{64.4}\right)\left(\frac{2.62\,I\,(0.15 + (11/CO_2))}{A \times 10^6}\,\frac{T_2}{T_1}\right)^2 . \quad \text{(A-13b)}$$

P = 30 in. Hg.
T_1 = 520 F, absolute.
F = 0.2 velocity heads (Fig. 2—vertical).
F = 1.4 velocity heads (Fig. 1—connector used).
CO_2 = 4 and 8 percent on dry (water condensed) basis.
T_2 = 520 + temperature rise above ambient
I/A, L/D as indicated in Tables.

Table 4: Total draft losses, in inches H_2O.

$$\text{Losses} = \frac{0.255\,P}{T_2}\left(\frac{F + 0.03L/D}{64.4}\right)\left(\frac{2.62\,I\,(0.15 + (11/CO_2))}{A \times 10^6}\,\frac{T_2}{T_1}\right)^2 \ldots \text{(A-13c)}$$

P = 30 in. Hg.
T_1 = 520 F, absolute.
F = 1.4 velocity heads (Fig. 2—vertical).
F = 2.4 velocity heads (Fig. 1—connector used).
CO_2 = 4 percent on dry (water condensed) basis.
T_2 = 520 + temperature rise above ambient
I/A, L/D as indicated in Table.

Table 6: Draft loss relative to losses at 300 F deg rise.

$$\text{If } T_2 = 300\,F + 520\,F = 820\,F$$

$$\text{Losses at } T_3 = (\text{Loss at } T_2) \times \frac{T_3}{820}$$

Table 7: Draft loss relative to losses at 4 percent CO_2.

$$\text{Loss at (other } CO_2) = \text{Loss at 4 percent } CO_2 \times \frac{(0.15 + (11/CO_2))^2}{(0.15 + (11/4))^2}$$

POINT-BY-POINT APPLICATION OF THE DRAFT EQUATION

Application of the draft equation to a gas venting system having a draft hood is essentially a 2-step process. The first step consists of establishing flow velocity, V, by

TABLE A-2—EVALUATION OF GENERAL DRAFT EQUATION AS APPLIED TO THE GAS VENT SHOWN IN FIG. A-1

$$S = 0.0075\ (0.96H - 7.12R)$$

CENTERLINE DISTANCE, L, FT., FROM OUTLET	HEIGHT TO OUTLET, H FEET	RESISTANCE R_s, VELOCITY HEADS	AVAILABLE STATIC DRAFT S, INCHES H_2O
0	0	0	0.0
5	5	0.15	−0.028
10	10	0.30	−0.056
15	15	0.45	−0.084
19	19	0.57	−0.105
Ell 20	20	1.35	−0.072
21	20	1.38	−0.0705
30	20	1.65	−0.0555
35	20	1.80	−0.048
40	20	1.95	−0.0398
45	20	2.10	−0.0315
50	20	2.25	−0.024
55	20	2.40	−0.0157
59	20	2.52	−0.0098
Ell 60	20	3.30	+0.0323
61	21	3.33	+0.027
65	25	3.45	+0.0045
70	30	3.6	−0.024
75	35	3.75	−0.052
Inside 80	40	3.9 (Max. R_s)	−0.080
Outside 80	40	5.4 (R_v)	0.0

use of methods in reference 3, or by applying Equation A-6 using total system resistance including draft entrance losses. Determination of static draft in gas vents requires that the sum of draft hood entrance loss and acceleration head (about 1.4 to 1.5 velocity heads) must be subtracted from the value of R_v used in Equation A-6 in order to obtain available static draft at the draft hood outlet.

The difference between R_v and R_s at the draft hood outlet expressed mathematically for this case is:

$$R_s = R_v - 1.5 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{(A-14)}$$

For any venting system, as will be shown presently, R_s varies with position in the system, while R_v is a system constant for a gas vent.

Consider a 12-in. diameter gas vent having a 12-in. size vertical draft hood, 20 ft of vertical piping, a 90 deg elbow, 40 ft of lateral piping, another 90 deg elbow, and 20 ft of additional vertical piping to make a total height of 40 ft. This vent is shown in Fig. A-1.

The following assumptions will be made concerning this vent.

(*a*) *Isothermal operation at 500 F deg above ambient, or negligible cooling.*

(*b*) *All fitting resistance is localized in the fittings.*

(*c*) *Ambient temperature T_1 is 520 F absolute.*

(*d*) *Resistances are as follows: acceleration head is 1.0 velocity head; entrance loss in draft hood is 0.5 velocity head; pipe friction is 0.03 L/D velocity head; 90 deg elbows (two used), at 0.75 velocity head each; outlet resistance is negligible; total resistance* $R_v = 3.0 + 0.03\ L/D = 3.0 + 0.03\ 0.80/1 = 5.4$ *velocity heads.*

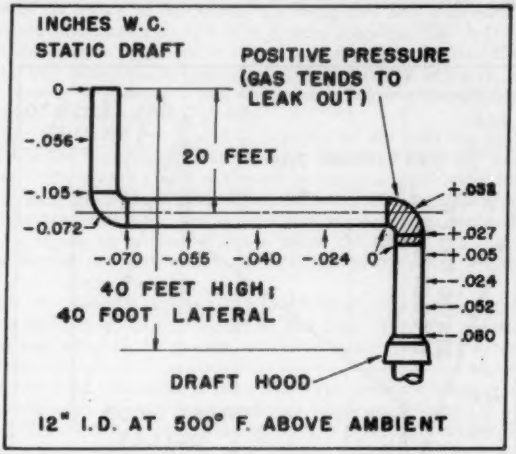


FIG. A-1—STATIC PRESSURES IN A VENTING SYSTEM COMPUTED USING EQUATION A-9

Values to substitute into Equation A-6 are: $H = 40$ *ft;* $T_2 = 1020$ *F, absolute;* $T_1 = 520$ *F, absolute;* $R_v = 5.4$ *velocity heads;* $2g = 64.4$ *feet per sec per sec.*

Solving for flow velocity in the vent:

$$V = \sqrt{\frac{2g\,H}{R_v}\left(\frac{T_2}{T_1} - 1\right)} \quad \ldots\ldots\ldots \quad \text{(A-15)}$$

$$= \sqrt{\frac{64.4 \times 40}{5.4}\left(\frac{1020}{520} - 1\right)}$$

$V = 21.40$ *ft per sec for the vent gases at a temperature of 500 F deg above an ambient of 60 F.*

To compute available static draft at the draft hood outlet, the resistance of the system between the draft hood and the outlet must be found using Equation A-14 or by summation of the resistance starting at the chimney outlet.

$R_a = 5.4 - 1.5 = 3.9$ velocity heads.

Solving Equation A-9 at a barometric pressure of 30 in. Hg:

$$S = \frac{0.255 \times 30}{1020}\left[40\left(\frac{1020}{520} - 1\right) - \frac{3.9 \times (21.40)^2}{64.4}\right]$$

$S = 0.080$ inches H_2O available static draft in the gas vent, at the draft hood outlet.

Emphasis must be placed on the important distinction between available static draft within a gas vent, which may be positive, negative or zero, and the fact that the vent itself produces zero available static draft at the draft hood inlet regardless of vent action, due to the highly effective draft-negating action of the draft hood. Occasionally there is a slight interaction between the appliance and its vent but it is usually so inconsequential that both appliance and vent design procedures can ignore it completely,

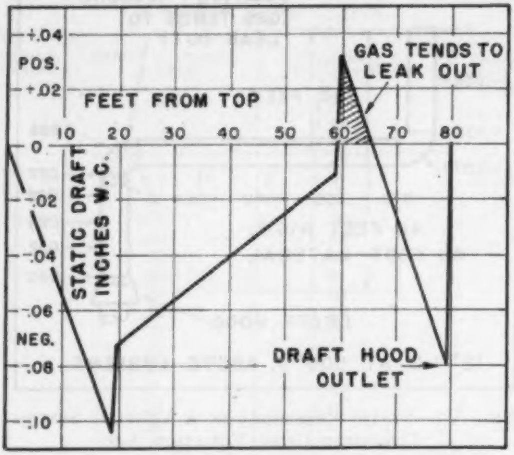


FIG. A-2—VARIATION IN SYSTEM STATIC PRESSURE WITH DISTANCE FROM VENT OUTLET (SEE FIG. A-1 FOR DIAGRAM)

providing the draft hood fulfills existing *American Standards Association* requirements. Gas vents, as has been frequently emphasized, must be designed for quantity of flow, therefore the draft existing within them is seldom relevant to their proper operation. For example, an undersized draft hood and vent attached to an appliance which produces more gases than the vent can carry away, conceivably might indicate a satisfactory available static draft figure, while causing simultaneous severe draft hood spillage. Accordingly the significance of static draft measurements in actual systems can be understood only by making a point by point analysis, in which pertinent information about ambient pressures caused by such factors as air supply resistance is also properly accounted for.

In the previously described gas vent, for which a velocity of 21.4 fps was computed, the available static draft at the very top is zero. The buoyant effect of the hot gases above the outlet must be neglected, because it cannot readily be included. The energy employed to accelerate the flowing stream to vent velocity is dissipated in end loss or

atmospheric turbulence, but by definition, this energy is not included in available static draft.

Applying Equation A-9 to the top or outlet of the vent, it can be seen that both H and R are zero at this level, thus S, the available static draft is also zero. Five feet below the outlet, some available draft exists. Resistance is 0.03 x 5 = 0.15 velocity heads and at this level:

$$S = \frac{0.255 \times 30}{1020}\left[5\left(\frac{1020}{520} - 1\right) - \frac{0.15\,(21.40)^2}{64.4}\right]$$

S = 0.056 in. H_2O, 5 ft below the vent outlet.

Static drafts at all points in the system are computed and tabulated in Table A-2. Selected values are also shown on the vent system diagram, Fig. A-1. The complete draft curve for this system is plotted in Fig. A-2, using distance from the outlet as the horizontal scale, and inches water column, either positive or negative, as the vertical scale. Where computed pressure inside the vent is below atmospheric, the negative pressure convention is employed in Fig. A-2.

As can be seen from Figs. A-1 and A-2, starting at the vent top and going down, static draft within the vent becomes increasingly negative until the elbow below the outlet is reached. Resistance losses of this elbow cause a rapid reduction in available static draft, as does the lateral itself, which adds friction without changing height. At the elbow over the draft hood, static draft reaches zero then becomes greater than atmospheric to continue for a short distance below the elbow. In this region, cross-hatched in Figs. A-1 and A-2, gases will tend to leak out of the vent. At all other points, leakage will tend to be into the vent.

It is of interest that use of 2 elbows and a long lateral reduces the system static pressure or available static draft to 0.08 inches or less than ⅓ of the theoretical value of 0.29 inches. At one point in the system, available static draft is greater than at the draft hood outlet.

Fig. A-2 indicates how ridiculous it is to use static pressure measurement within a gas vent as an indication of system performance, since static pressure literally varies *all over the map*. On the other hand, since velocity is practically constant at all points within this or any other individual gas vent system, velocity, volume flow or dilution ratio measurements provide an absolutely positive criterion of vent operation.

Where available static draft at the appliance is to be controlled by a barometric damper, Fig. A-1 shows that it is essential to locate the control at the appliance, rather than at some more convenient place in the system. For example, had a barometric damper been located in the vicinity of the elbow just over the draft hood outlet, it would be operating in a region of positive pressure, and either would be unable to control, or might lead the installer to blame a non-existent downdraft for his troubles.

More precise analysis of this system under actual rather than ideal conditions may be obtained by use of the Kinkead equations[8], with their precise approach to heat loss, however for vent or stack diameters greater than 8 inches, the effect of gas cooling has a diminishing effect on draft and flow within the system.

DISCUSSION

W. B. KIRK, Cleveland, Ohio, (WRITTEN): Mr. Stone has made an outstanding contribution to heating and air conditioning by reducing theoretically sound but complex venting formulas and design procedures to a simple and readily usable set of tools. To do this required tedious evaluation of many simplifying assumptions as well as a talent for making easy interpretations. The simplified design procedure set forth in the paper appears to be valid for single unit installations.

There are some limitations in the application of this procedure which are implied by Mr. Stone, but should be emphasized. For vent systems with horizontal flue connec-

tors, the tables are based upon the smaller flue area, regardless of whether it is in the connector or vertical stack. In both of the prepared examples, the vertical stack diameter could be doubled with no change in the evaluated stack height as long as the diameters of horizontal flue connectors remained 60 and 23 inches, respectively. Mr. Stone assumes the most economical chimney will be one with a diameter approximately equal to the diameter of the appliance flue connector. This assumption may be obvious for the large gas appliances (22,000,000 Btu per hour and 6,000,000 Btu per hour inputs) employed in the examples. For smaller appliances, however, it may be more economical to vent several appliances into a central chimney rather than construct individual chimneys.

Venting of two or more gas appliances equipped with barometric dampers or draft hoods into a single chimney introduces dilution air effects which are not covered in this paper. When dilution air is entrained through the barometric damper or draft hood relief opening of a non-operating appliance, the temperature of the flue gases produced by the operating appliances is lowered, thereby changing the chimney requirements of the multiple vent system as compared to those for a single appliance.

The lowering of flue gas temperatures by entrained dilution air may also occur if several high input gas appliances, each employing a draft hood, are connected to a common manifold which then enters the chimney as a single flue connector. If all the appliances are not operated simultaneously, then the chimney must be designed for that operating condition which gives the lowest flue gas temperature in the common manifold.

Under sponsorship of the *American Gas Association*, research is being conducted at the Association's Laboratories on venting several gas appliances into a common chimney. While this study is restricted primarily to residential vent systems, the basic techniques developed should apply to large diameter chimneys and flue connectors. Mr. Stone's company has also developed useful information along these lines.

C. L. Benn, Pittsburgh, Pa., (Written): It is my desire to compliment Mr. Stone for the presentation of this excellent paper. Representing, as it does, the results of research, study, and correlation of data, it is difficult to comment without having been directly involved in research on this subject.

However, after carefully studying the paper, I present the following comments in the hope that they will be a small contribution to this important design problem.

Two of the most important factors involved in the design of a gas appliance, gas furnace or gas boiler installation, is the design of a proper chimney or stack, and the provision for admission of air for combustion, make-up air and ventilation of boiler and furnace rooms. The second factor is not covered in this paper, but I mention it as being essential to venting, or as a correlated problem in connection with chimney and stack design.

The information presented in this paper closely checks with *American Gas Association* research on the subject. For example, referring to the problem of determining the height and diameter of stack required for a gas furnace having an input of 22,000,000 Btu per hour, the temperature of the products of combustion entering the draft hood is given as 380 F. The partial flue gas analysis given in the problem is within that found in actual installations as indicated by the *A.G.A. Field Survey of Gas Appliance Venting Conditions*, Research Report No. 1243, February 1956, on which Mr. Stone was a Technical Advisor. This research report indicated that vent connector flue gas temperatures at the outlet of the draft hood averaged 238 F, and inlet temperatures averaged 412 F.

The statement is made in the paper that *temperature in the boiler room does not influence chimney operation. If the chimney design is adequate for 60 F operation, any drop in outdoor temperature will improve stack action*, . . . It seems to me that this statement should be qualified, since, in a given stack with temperature of flue gas of say 238 F, a drop in outdoor temperature will tend to decrease the temperature of the flue gases in the stack, and decrease the stack action. Quoting from the *Vent Installation Handbook* by the Metalbestos Division of Mr. Stone's Company, the statement is made that

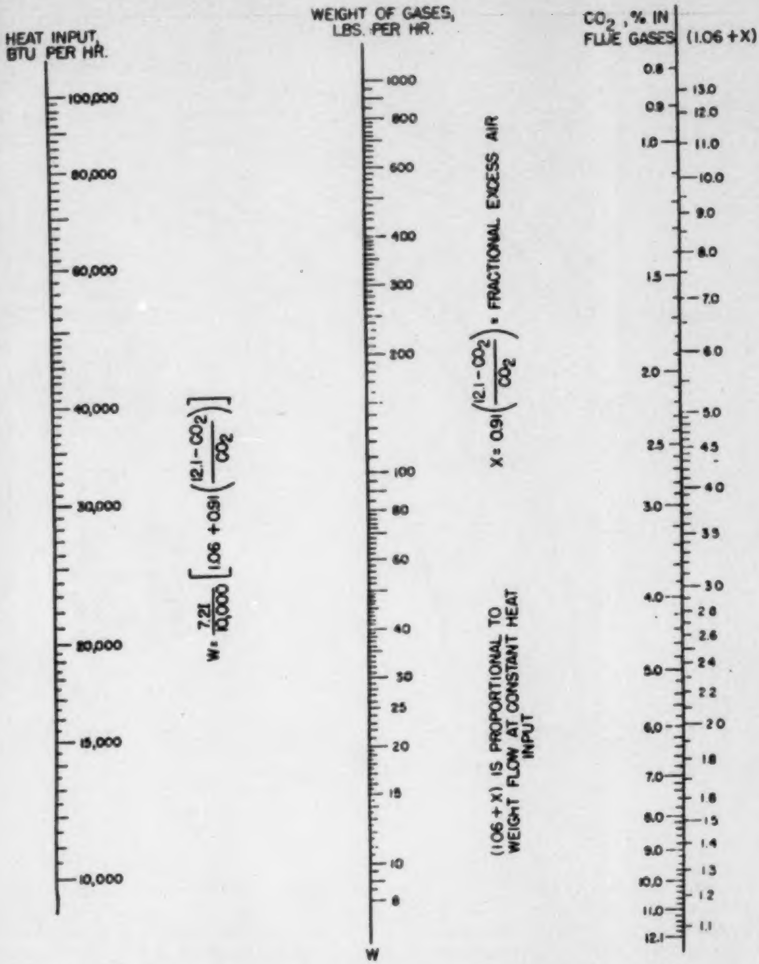


FIG. A—NOMOGRAPH FOR WEIGHT FLOW OF FLUE GASES, NATURAL GAS

the operation of a gravity vent depends on having the vent gases inside the vent hotter (and therefore lighter) than the surrounding air. The hotter the vent gases, the lighter they are and the greater is the force that expels them up through the vent. I would like Mr. Stone's comment on this point.

Finally I would like to present a few brief comments with regard to the related problem of providing means of admission of air for combustion, make-up air and ventilation

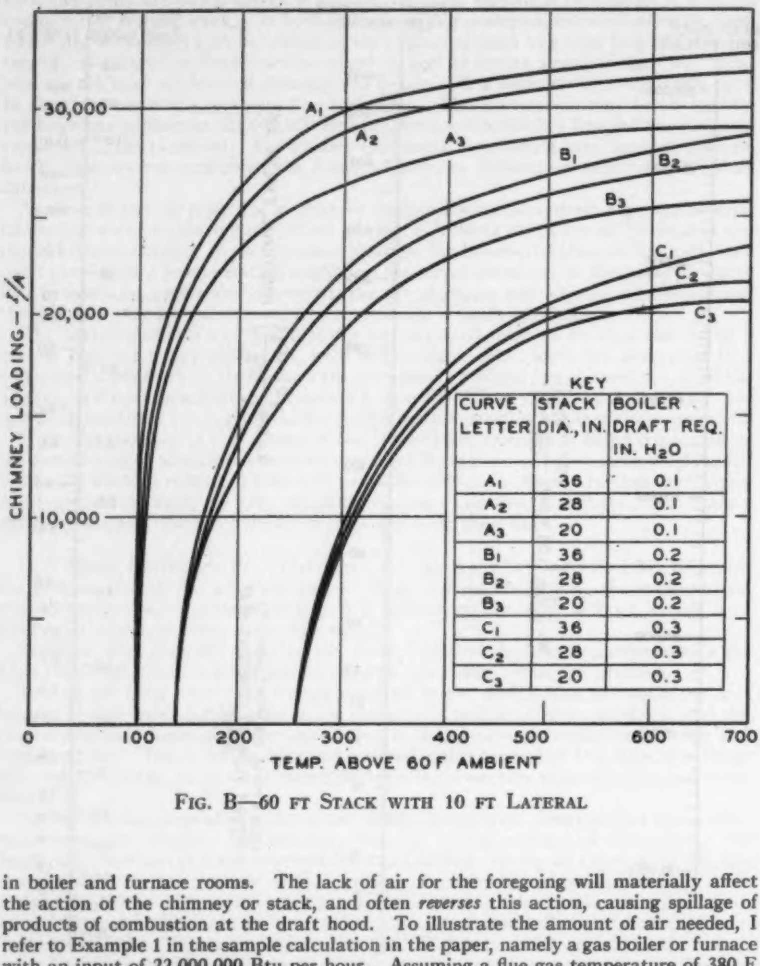


FIG. B—60 FT STACK WITH 10 FT LATERAL

in boiler and furnace rooms. The lack of air for the foregoing will materially affect the action of the chimney or stack, and often *reverses* this action, causing spillage of products of combustion at the draft hood. To illustrate the amount of air needed, I refer to Example 1 in the sample calculation in the paper, namely a gas boiler or furnace with an input of 22,000,000 Btu per hour. Assuming a flue gas temperature of 380 F entering the draft hood, a flue gas analysis of 8 percent CO_2, 30 percent excess air, 6 percent O_2, and 1020 Btu per cu ft gas.

Again, referring to the *A.G.A.* Field Survey of Gas Appliance Venting Conditions, the draft hood dilution effect, namely the ratio of the weight of flue gases flowing in the vent connector to the appliance flue gas weight flow, averaged 2.16; that is, 1.16 pounds of dilution air drawn through the draft hood relief opening for each pound of flue gases entering the draft hood.

Referring to Fig. A which is the same as Fig. 104, from the *A.G.A.* Research Bulletin 74, *Principles of Draft Hood Operation and Design*: At 8 percent CO_2, the weight of flue gases per 100,000 Btu per hour input is 109 pounds.

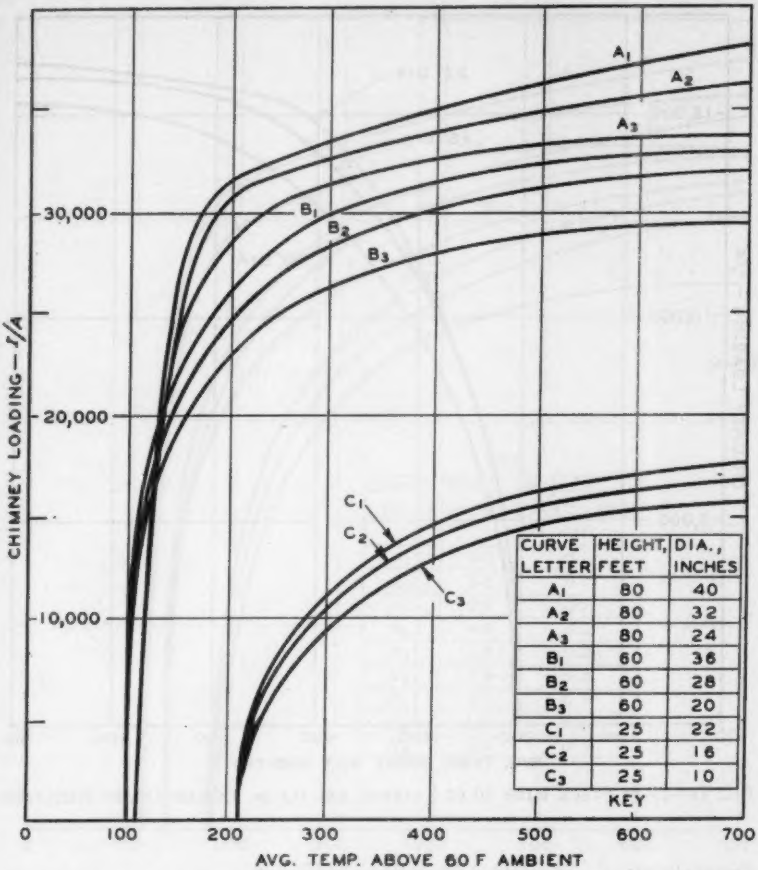


Fig. C—Various Chimneys with 10 ft Laterals and 0.1 in. Boiler Draft Required

On this basis, a boiler or furnace with an input of 22,000,000 Btu per hour would generate 220 x 109 = 23,980 pounds of products of combustion. The pounds of dilution air entering the draft hood would be 23,980 x 1.16 = 27,817. For approximation, assume this to be standard air with a density of 0.075 pounds per cu ft.

$$\frac{27{,}817}{0.075} = 370{,}893 \text{ cu ft per hour of dilution air.}$$

Since 10 cu ft of air per cu ft of gas is required for complete combustion, and the example states 30 percent excess air, the air drawn into the burners will be 13 x 21569 (cu ft of 1020 Btu gas) = 280,397 cu ft per hour.

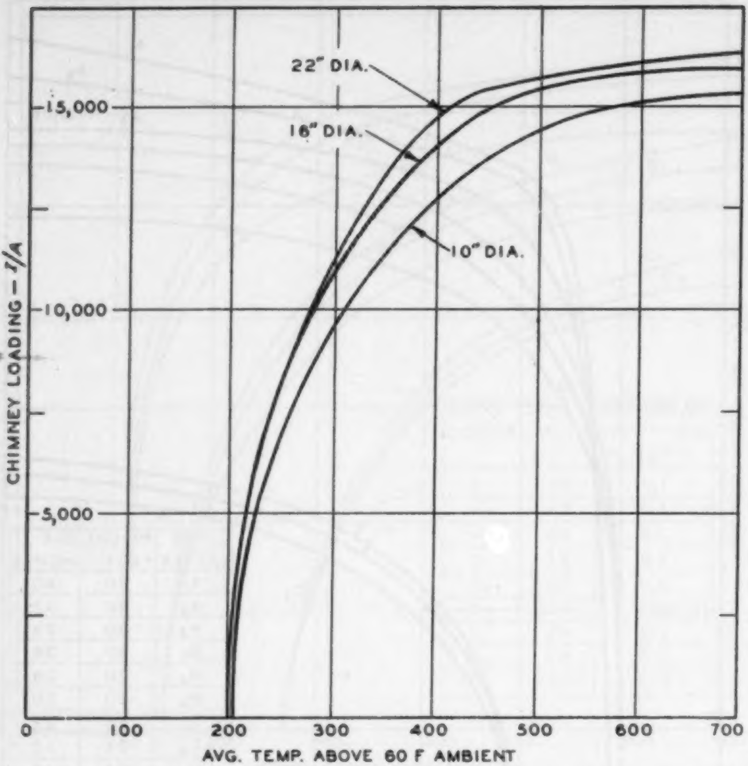


Fig. D—25 ft Stack with 10 ft Lateral and 0.1 in. Boiler Draft Required

Recapitulating:

Dilution air = 370,890 cu ft per hour.
Air for combustion = *280,397* cu ft per hour.
Total make up air for room = 651,287 cu ft per hour.

Architects have a penchant for designing buildings with no provisions for admission of air for combustion and make-up air into boiler and furnace rooms.

The immediate foregoing is cited to illustrate the problem and to emphasize that it is a factor in chimney and stack design.

C. G. Segeler*, New York, N. Y. (Written): My comments on Mr. Stone's paper are necessarily brief because he and I have worked for nearly a year on this subject in connection with his contribution to the proposed *Gas Engineers' Handbook*. These

* Director, Utilization Bureau, American Gas Association.

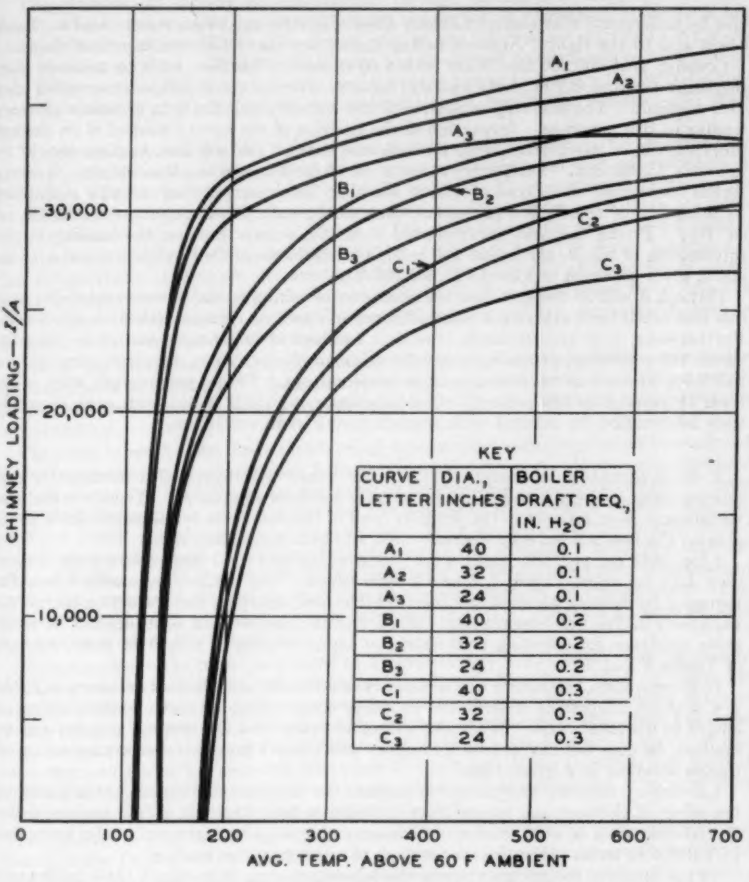


FIG. E—80 FT STACK WITH 10 FT LATERAL

comments are that the method furnishes a rational basis for sizing; and that the stacks are likely to be smaller than shown in the typical boiler manufacturer's catalog.

Because of the several steps required to use the proposed method, I felt that some selected curves could be drawn based on specific stack dimensions to simplify selection of sizes and capacities. Such curves would not cover all cases. Nevertheless, they make it possible to secure an almost immediate answer just by inspection. Figs. B, C, D and E are a set of these curves and illustrate how the Stone method could be extended.

These curves are intended to show suitable chimney sizes when attached to boilers which require 0.1, 0.2 or 0.3 inches of draft. To show how these charts work, it should

first be understood that correct answers must either be on the curves or close to them, below and to the right. Answers falling above any curve indicate incorrect sizing.

Consider a 15,000,000 Btu boiler with a 60 ft stack. Further, let it be assumed that the boiler requires 0.2 in. draft and will have an average stack temperature 400 F deg over ambient. The first step to determine the correct stack size is to assume a *chimney loading* in Btu per sq in. Inspection of the position of the curves marked B on the appropriate chart (60-ft stack, Fig. B) indicates that at the 400 line, loading should be less than 25,000 Btu. I might try to see if the 20-in. diameter stack would do. Twenty inches correspond to an area of a little less than 300 square inches and this multiplied by a loading of 20,000 Btu per square inch would only permit inputs of 6,000,000, or so, Btu. Trying the next curve proves to be the answer because the loading at the intersection of the B_2 curve and 400 is 24,000 which, multiplied by the area of a 28-in. stack, gives an input of exactly 15,000,000 Btu.

Perhaps it will be thought that the charts do not simplify the job very much because one still needs both experience and judgment in handling chimney selection effectively. Furthermore, it is unfortunately true that catalogs of boiler manufacturers (sizes between 100 to 500 hp) generally do not furnish enough data as to the draft required for gas-firing at various percentages of nominal ratings. These percentages may range from 75 percent to 150 percent. Our experience is that it is not even easy to secure such information by contact with manufacturers' representatives.

P. R. ACHENBACH, Washington, D. C.: Mr. Stone has suggested a method for correlating chimney height and chimney area for gas-fired equipment. These are the two variables of most interest to the designer; and of the two there is less information available on the relation between chimney area and heat generation rate.

I feel that the author's paper would be strengthened if he had reported the laboratory data on which Tables 2, 3 and 4 were based. The reader can readily follow the sequence for moving through the several tables and applying the correction factors for variations in flue gas temperature, carbon dioxide content, and altitude, but he is not given guidance for selecting trial values of input loading or length to diameter ratio in Tables 2 and 3.

In his examples, the author has arbitrarily selected certain points of entrance to Tables 2–4 without suggesting a basis for choosing these values of input loading factor or length to diameter ratio. While the author has described the method as a cut-and-try method, he does not reveal how a designer will know when he is converging on an optimum solution in a given case.

Laboratory data are also needed to support the recommendations in Table 6 showing the effect of chimney gas temperature on relative draft loss. It is also suggested that the cooling effect in steel stacks and masonry chimneys be expressed in the lower part of Table 6 in terms other than percentage of a thermometer reading.

In my opinion, the presentation of the laboratory data from which these tables were derived, and some guidance as to the choice of entrance values for design purposes would make Mr. Stone's paper a valuable and authoritative contribution to the literature on chimney design.

AUTHOR'S CLOSURE: I want to thank Mr. Kirk, Mr. Benn, Mr. Segeler and Mr. Achenbach for their comments and penetrating analysis of my paper. There is every indication that much further work needs to be done in the field of chimney design to provide solutions to the problems these gentlemen have raised.

Mr. Kirk observes that the problem of differing connector and stack areas is handled by using the smaller area for determining losses. Because pressure drop is proportional to the square of velocity, it can readily be calculated that doubling the area of one part of the system causes a very small drop in overall losses. The use of oversized vertical stacks to anticipate future needs is desirable but this will have no effect on these design computations.

The problems of varying air entrainment through several draft regulators or draft hoods, where two or more gas appliances are connected to a common chimney is not covered by the paper. There is no simple way to use information provided for the design of these systems. While our company has developed comprehensive data on system design with multiple draft hoods, we have only begun to explore the more complicated problem of draft regulators, or combinations of barometric dampers and draft hoods.

The curves prepared by Mr. Segeler pose an interesting question. Considering ease of interpolation and analysis, curves have numerous advantages over tabulated data. However, when I first became involved with problems of presenting information for handbook use, it was pointed out to me that the typical user is far more likely to understand tabulated data, than a set of curves. The draft loss tables comprise a 5 dimensional system involving the following variables, (1) Draft loss, (2) Input loading, (3) Gas temperature, (4) Length to diameter ratio, (5) CO_2. To plot all these would necessitate multiple sets of curves or the use of grid-charts, which are likely to confuse the casual user. While curves are usually preferable, the tabular presentation here is thought to be more usable and concise.

One of the problems in chimney diameter selection is that there is no guide to the original choice of diameter. Thus selection of diameters and drafts for plotting curves limits their application to those selected variables, while the tabulated method retains full flexibility.

The point raised by Mr. Benn on effects of outdoor air temperature must be analyzed from the standpoint of steady-state boiler or furnace operation. If steam is produced at a given rate and pressure, boiler flue product temperature remains constant. A drop in outdoor temperature will thus increase the temperature difference between the flue products inside the chimney and outdoor air. With an adequate air supply to the boiler room, under these conditions theoretical draft will increase. In large stacks the effect of cooling due to heat loss at low ambient temperatures is more than compensated for by the increase in theoretical draft due to lowered ambient air temperature. The Kinkead equations of vent operation as presented in *A.G.A. Research Bulletin 68*, can be used to analyze this situation.

The appendix to the paper as printed in TRANSACTIONS covers briefly how air supply resistance can be calculated. I agree completely with Mr. Benn that adequate air supply is vital to adequate chimney capacity. Although much study has been devoted to this essential detail, rules for air supply for large gas-burning equipment have not been formalized as readily as for domestic installations. In general, however, it appears that an opening having a free area equal to stack area, when augmented by normal infiltration, should provide for minimum replacement air requirements, providing that there are no interfering ventilating or exhaust systems.

In response to Mr. Achenbach's questions, I feel certain that the appendix material printed in the TRANSACTIONS shows that the design tables possess an adequate theoretical foundation. As a matter of interest, the equations were subjected to a laboratory check, with computed available static draft being compared with observed available static. Allowing for the usual experimental discrepancies resulting from the use of a typically recalcitrant micromanometer the results indicated a satisfactory degree of correlation.

The tables themselves are simply tabulated point solutions to the equations for chimney section, and can readily be checked by substitution in the appropriate equation. Their value lies in the fact that they permit simple, rapid solution to chimney design problems by a convergent method. I cannot provide a precise explanation as to why it is possible to make a wild first approximation and still end up with a usable answer in two or three steps.

However, to answer Mr. Achenbach's question on criteria for flue gas velocity, I can only say that this is one factor with which the designer need not be concerned. In effect this is capacity design method rather than a velocity design method. If the stack as designed produces the proper draft, and does not have excessive pressure drop, gas

velocity can vary over a wide range, depending on whether the stack is short or very high. The design method adjusts the velocity to provide the correct capacity.

Table 6, relating temperature to relative draft loss, is a simple conversion from gas temperature to volume to velocity squared. It can be checked by analysis of Equations 12 and 13 of the appendix. If Equation 13 is multiplied out it can be seen that the losses vary with the first power of T_2, thus at 1000 F deg gas temperature rise, the loss will be $\frac{1000 + 520}{300 + 520}$ times the loss at 300 F or 1.85 times as great.

I freely admit that expressing gas temperature drop as a percentage has little real significance. By comparison with some of the preferred methods such as the solution of the Kinkead equations, it has practicality and simplicity to recommend it. Having explored this problem from both theoretical and practical points of view, I have decided that almost any sort of educated guess will provide adequate heat loss compensation. Also I have found that if stack diameter exceeds 12 inches, at practical flow rates the effect of cooling can either be neglected or readily handled by designing for slightly more draft than necessary.

1637

No. 1637

PRESSURE LOSSES AND FLOW CHARACTERISTICS OF MULTIPLE-LEAF DAMPERS†

By Edward J. Brown* and Julian R. Fellows**, Urbana, Ill.

THE OBJECTS of this investigation were: (1) to determine the resistance to air flow of multiple-leaf dampers for all leaf positions from wide open to closed; (2) to determine the effect of damper resistance on the control of air flow through a duct system; (3) to determine the effect of total system resistance on the flow-control characteristic of any particular damper.

Description of Apparatus

The information presented herewith pertains to the parallel- and opposed-leaf dampers listed in Table 1. The damper frames were ¼ in. less on each dimension than the sizes listed in Table 1. The dimensions listed are those of the ducts in which the dampers were installed. The damper frames were constructed of ¼-in. flat-bar stock. The damper leaves in every case were 1 in. less in length than the first dimension listed for the damper size. All dampers were equipped with ⅜-in. wide stop strips to fill the gap between the leaves and the frame. All of the leaves had crimped edges. The depth of the crimp was the same for all leaf heights.

An elevation view of the fan and ductwork is shown in Fig. 2. The constant-speed fan was capable of delivering 1000 cfm of standard air against a static pressure of 7.5 in. water or 15,500 cfm of standard air against a static pressure of 0 in. water. The fan was directly connected to a 10 hp electric motor. The static pressure at Station 1 was adjusted by controlling, with the face and bypass damper, the volume of air entering the test duct. The face and bypass damper was operated by a reversible electric motor which could be stopped in any position. The switch for the control of this motor was located at the central gage board.

Flow straighteners of the tubular type, 3 in. in diameter and 10 in. in length, were included in the test arrangement. The air-flow measuring section consisted of a short converging entrance, a 14-in. diameter throat section which contained a fixed pitot-static tube located in the center of the throat section, and a long di-

† This article includes the results in part of an investigation conducted in the Mechanical Engineering Laboratory of the University of Illinois under the terms of a cooperative agreement between the University and the Johnson Service Co.

* Research Assistant in Mechanical Engineering, University of Illinois.
** Professor of Mechanical Engineering, University of Illinois. Member of ASHAE.

Presented at the 64th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Pittsburgh, January 1958

verging section. The stationary pitot-static tube was calibrated by a 20-point pitot-static tube traverse at Station 2. Calibration tests were conducted at several flow rates and a calibration curve giving weight of standard air in pounds per hour plotted against velocity pressure at the fixed pitot-static tube was prepared from the data obtained. Velocity pressures at the measuring section were obtained

TABLE 1—DAMPERS INCLUDED IN THE TEST PROGRAM

DAMPER SIZE, INCHES	TYPE[a]	DAMPER[b] AXIS	NUMBER OF LEAVES	LEAF HEIGHT, INCHES	LEAF LENGTH, INCHES	PERIMETER, INCHES	TOTAL LEAF LENGTH TO PERIMETER RATIO (L/R)
12x24	P	V	3	8½	11	72	0.46
12x32	P	V	4	8½	11	88	0.50
12x40	P	V	5	8½	11	104	0.53
12x48	P	V	6	8½	11	120	0.55
12x48	P	V	8	6½	11	120	0.73
12x48	P	V	4	12¼	11	120	0.37
12x36	P	H	4	9¼	11	96	0.46
24x36	P	H	4	9¼	23	120	0.77
36x36	P	H	4	9¼	35	144	0.98
48x36	P	H	4	9¼	47	168	1.12
24x36[c]	P	H	4	9½	20¾	120	0.69
12x32	O	V	4	8½	11	88	0.50
12x48	O	V	6	8½	11	120	0.55
12x48	O	V	8	6½	11	120	0.73
12x48	O	V	4	12¼	11	120	0.37
12x36	O	H	4	9¼	11	96	0.46
24x36	O	H	4	9¼	23	120	0.77
36x36	O	H	4	9¼	35	144	0.98
48x36	O	H	4	9¼	47	168	1.12
24x36[c]	O	H	4	9½	20¾	120	0.69
30x48[c]	O	H	5	9¾	27⅞	156	0.89

[a] P = Parallel leaf; O = opposed leaf.
[b] H = horizontal axis; V = vertical axis.
[c] Dampers not manufactured by sponsor.

with a micromanometer which could be read directly in increments as small as 0.001 in. water.

The duct section in which the test dampers were installed consisted of nine 44-in. sections of 3 x 4 ft duct mounted on movable stands. Each duct section was provided with 2-in. flanged joints. The face of one flange in each case was provided with a soft rubber gasket. This gasket was compressed when the duct sections were bolted together and prevented leakage of air at the duct joints. Four sections of this duct were on the upstream side and 5 sections were on the downstream side of the test damper. A baffle consisting of 4 layers of 16-mesh screen wire was placed upstream from the test damper to produce a more uniform air velocity in the duct immediately upstream from the face of the test damper.

The test dampers were bolted to the duct wall at the location indicated in Fig. 2. The joint between the damper frame and the duct wall was sealed with plastic tape. Horizontal and vertical partitions were provided in the 3 $\times$ 4 ft

duct when needed to make a full-length duct the same size as any damper which was smaller in either length or height than the dimensions previously mentioned. This eliminated the necessity of providing a different duct for each damper size.

Piezometer rings located at Stations 3 and 4 and connected to an inclined water manometer provided damper pressure loss data. Pitot-static tube traverses indicated that the air stream filled the duct at Station 4 and that the velocity was the

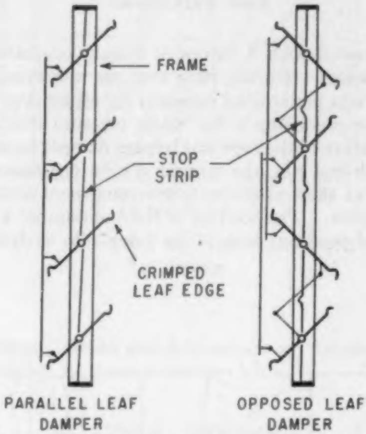


FIG. 1—TWO TYPES OF MULTIPLE-LEAF DAMPERS

GLOSSARY OF TERMS

Crimped edge—edge of the leaf parallel to the axis is provided with a crimp to allow the leaves to overlap when closed and to add strength to the leaf (see Fig. 1).

Damper position—position of the damper leaves with respect to the plane of the frame, 0 deg (closed), 90 deg (open).

Frame dimensions—dimension parallel to the leaf axis is given as the first dimension. The second dimension given is measured perpendicular to the leaf axis.

Horizontal or vertical axis—orientation of the leaf axis depending on whether the damper is to be mounted in the duct with the leaf axis horizontal or vertical.

Leaf height—height of a single leaf measured perpendicular to the leaf axis.

Leaf length to perimeter ratio—the sum of the lengths of all the leaves divided by the perimeter of the duct. Leaf length to perimeter ratio is abbreviated to L/R ratio.

Opposed-leaf damper—adjacent leaves rotate oppositely (see Fig. 1).

Parallel-leaf damper—all leaves remain parallel to one another as they are rotated (see Fig. 1).

Stop strips—strips of metal placed on the frame of the damper. The leaf ends and edges rest against the stop strips when the damper is closed. (see Fig. 1).

same as that at Station 3. Therefore, the total pressure loss between Stations 3 and 4 was assumed to be equal to the static-pressure loss between the same locations. The static-pressure loss was corrected for duct friction between Stations 3 and 4.

A second damper was installed at the end of the test duct to provide a means of varying the overall resistance of the duct system.

TEST PROCEDURE

Each damper was tested with 5 values of system resistance; namely, 1.5, 2.5, 3.5, 4.5, and 5.5 in. water. During each test run the system resistance (static pressure at Station 1) was maintained constant for all air flow rates, thereby simulating the use of a fan producing a flat static pressure characteristic. This was accomplished by operation of the face and bypass damper located just downstream from the fan. In each test run, the desired system resistance was established by means of the damper at the end of the test arrangement with the test damper in the 90 deg (open) position. The position of the end damper was maintained for all of the test-damper leaf positions from 90 to 0 deg. In each case, the test-damper

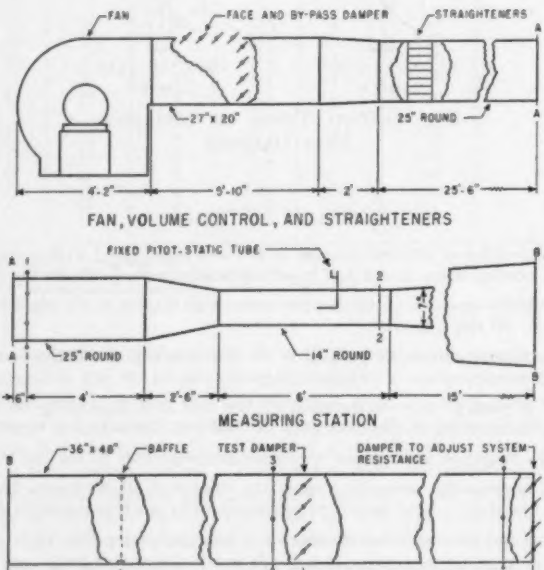


FIG. 2—ARRANGEMENT OF TEST EQUIPMENT

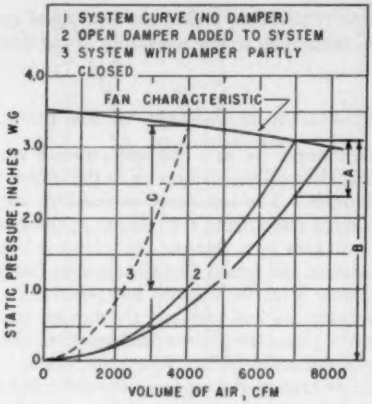


FIG. 3—CONTROL OF AIR FLOW WITH A DAMPER

leaf positions were changed by 10 deg increments as the test progressed. The required data were recorded for each position.

FAN, SYSTEM, AND DAMPER

Fig. 3 illustrates the relation between the static-pressure characteristic of a fan and the resistance characteristic of the duct system to which it is connected. The resistance of the duct system varies approximately as the square of the volume flowing through it. The intersection of Curve 1 and the fan characteristic curve indicates the volume of air that the fan will cause to flow through the duct system. Curve 2 represents the same system with the addition of an open damper. The resistance of the open damper decreases the air flow as indicated by the intersection of the fan characteristic curve and Curve 2. As will be explained later, the ratio of the wide open damper resistance, A, to the system resistance including the damper, B, is important in determining the ability of the damper to control the flow of air through the system in a satisfactory manner. The ratio of A to B is constant as long as the system remains constant, regardless of the volume of air handled. Thus, A/B with 4000 cfm flowing through the system is equal to A/B with 7000 cfm flowing through the same system.

Partially closing the damper modifies the system and reduces the air flow. Note the intersection of Curve 3 and the fan characteristic curve in Fig. 3. The partially closed damper adds resistance to the system equal to the difference, C, between Curves 2 and 3.

To decrease the flow of air through a duct system, resistance such as C in Fig. 3 must be added by turning the damper leaves from their wide open position. The resistance which must be added to a system for any specified flow rate between maximum and minimum may be determined by deducting the system resistance at that flow rate from the static pressure produced by the fan at the same flow rate.

The damper leaf position required to create the additional resistance may be determined if information pertaining to the resistance of the damper at all leaf positions is available.

RESISTANCE OF DAMPERS TO AIR FLOW

The resistance of the damper for any one leaf position varies as the square of the volume of air passing through the damper or as the square of the velocity of the air approaching the damper. The resistance or pressure loss caused by any leaf position is dependent on the free area of the damper as compared to the duct cross-sectional area. The free area of a damper includes the open area between the leaves plus the area between the leaves and the damper frame. The ratio of free area to duct cross-sectional area, for a given leaf position, is not constant for all dampers. The area between the leaf ends and the damper frame is nearly constant regardless of leaf length. Thus, for a given leaf position, the ratio of free area to duct cross-sectional area decreases as leaf length increases.

Measurement of the free area of a damper is very difficult for leaf positions other than 90 deg. Therefore, the measured values of pressure loss for each leaf position were expressed as that loss which would be caused by an abrupt expansion of the air stream from an equivalent contracted area to the duct cross-sectional area.

The Borda formula[1] for the loss of total head resulting from an abrupt expansion is,

$$H_t = (V_1 - V_2)^2/2_g \quad \ldots \ldots \ldots \ldots \quad (1)$$

where H_t is the total pressure lost in feet of fluid flowing, V_1 is the velocity of the air stream in the smaller area, V_2 is the velocity in the larger area (see Fig. 4).

$$\text{Since } A_1V_1 = A_2V_2 \quad \ldots \ldots \ldots \ldots \quad (2)$$

where A_1 and A_2 are the small and large areas, respectively, the loss may be expressed in terms of the velocity in the large stream. It is convenient in this application to express the loss in this manner, since the velocity in the damper flow area is difficult to measure.

$$\text{Then } H_t = [(A_2/A_1) - 1]^2 \, (V_2^2/2_g) \quad \ldots \ldots \ldots \quad (3)$$

The nozzle area of Section 1, Fig. 4, represents the equivalent contracted area, and Section 2, the duct area downstream from the damper. The abrupt expansion is assumed to occur as the stream at Section 1 expands to the dimensions of the duct at Section 2. The loss is H_t, given by Equation 3.

If Equation 3 is rearranged in the form

$$[H_t/(V_2^2/2_g)] = [(A_2/A_1) - 1]^2$$

the loss is then expressed in terms of velocity heads referred to the velocity either at Section 2 or at Section B, Fig. 4.

Since H_t and V_2 were known from the test data, it was possible to determine either A_2/A_1 or its reciprocal A_1/A_2 for each leaf position for all dampers tested. A_1/A_2 is the equivalent contracted area ratio. From here on in this discussion, the contracted area ratio will be referred to as A_e/A where A_e is the equivalent contracted area caused by the damper and A is the duct cross-sectional area.

---

[1] Pressure Losses Resulting from Changes in Cross-Sectional Area in Air Ducts, by A. P. Kratz and J. R. Fellows (University of Illinois Engineering Experiment Station *Bulletin No.* 300).

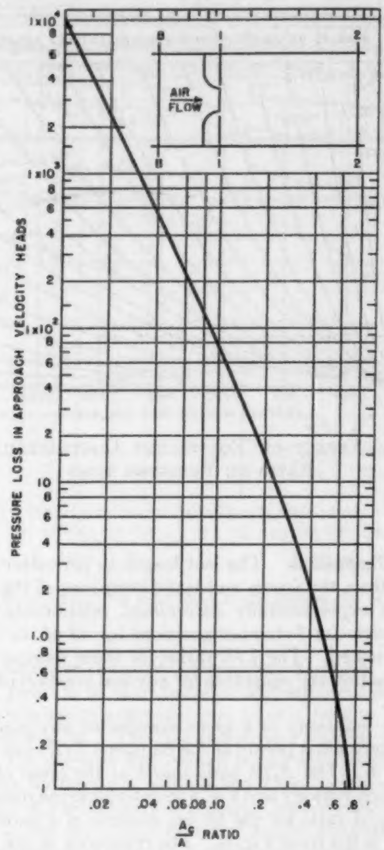


FIG. 4—PRESSURE LOSS DUE TO ABRUPT EXPANSION

A plot of the loss expressed as velocity heads referred to the velocity of the air approaching the damper, against the A_c/A ratio is given in Fig. 4. The curves of Fig. 5 were obtained from Fig. 4 by determining the loss for each of several A_c/A ratios over a range of approach velocities from 0 to 2800 fpm. The resulting losses expressed in terms of inches of water were plotted against approach velocity. Each A_c/A ratio corresponds to a leaf position. For instance, the 0.58 A_c/A ratio expresses the pressure loss or resistance of the wide open damper (leaves at the 90 deg position). To use the information presented in Fig. 5 for a particular damper, it is necessary to know the relation which exists between the A_c/A ratio

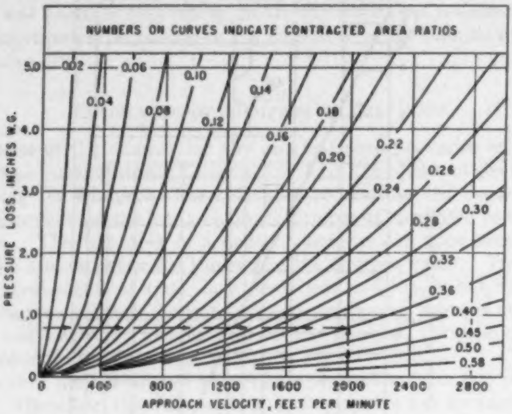


FIG. 5—EFFECT OF EQUIVALENT CONTRACTED AREA RATIO ON PRESSURE LOSS

and that damper's dimensions. The leaf length to perimeter (L/R) ratio may be readily determined from the frame and leaf dimensions of the damper.

Fig. 6 shows the experimentally determined relationship between the A_e/A ratio and the L/R ratio for 9 damper positions for all of the parallel-leaf and opposed-leaf dampers tested. The L/R ratios for these dampers are listed in Table 1. With this information the resistance for any leaf position of a particular damper may be determined.

To determine the resistance of a given damper at any position, it is necessary first to locate the intersection points of the damper's L/R ratio line and the damper position line in Fig. 6. The A_e/A ratio found at the point of intersection is then used with the approach velocity and Fig. 5 to determine the resistance of the damper. For example, the A_e/A ratio for the 50 deg position of a parallel-leaf damper with an L/R ratio of 0.8 is 0.3 from Fig. 6. The resistance of the damper at this position with an approach velocity of 2400 fpm is 2.0 in. water from Fig. 5.

The influence of the L/R ratio on the A_e/A ratio is illustrated by the slope of the curves in Fig. 6. That is, as the number of leaves in a given frame size is increased, the A_e/A ratio is decreased and the resistance of the damper is increased, which is to be expected. Also as the L/R ratio increases, the area at the end of the leaves becomes a lesser part of the total flow area and decreases the A_e/A ratio for a given leaf position.

The depth of the crimp at the leaf edges was constant for all leaf heights, with the result that the leaves of different heights were not geometrically similar. It was assumed that the lack of geometric similarity was of limited significance for the range of leaf heights included in the test results.

Fig. 6 indicates a greater A_e/A ratio for a given damper leaf position (with the exception of the 90 deg position) for parallel-leaf dampers as compared to opposed-leaf dampers. Thus, the pressure loss for a given velocity and leaf position is

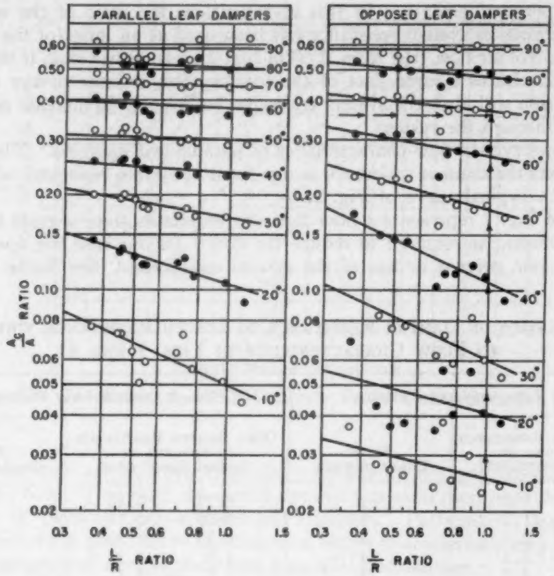


FIG. 6—EFFECT OF L/R RATIO ON THE A_c/A RATIO

less for parallel-leaf dampers than for opposed-leaf dampers. The average A_c/A ratio for the 90 deg position of both parallel- and opposed-leaf dampers is equal to 0.58 which results in a loss of approximately ½ of a velocity head referred to the approach velocity (see Fig. 4). This is a convenient number to use in system design calculations for the pressure loss caused by an open-multiple-leaf damper.

Fig. 6 includes data on dampers manufactured by 2 different companies other than the sponsor. All of the dampers tested were nearly identical in leaf construction. Therefore, it is not surprising that the results obtained were similar. It appears that the data presented may be applied to any multiple-leaf damper similar to those of Fig. 1.

TYPICAL FLOW CHARACTERISTIC CURVES

The flow characteristic of any damper-system combination is the relation between the percent of air flowing and the leaf position of the damper. It is a graphical representation of the damper's ability to control air flow. The flow characteristic is dependent on the ratio of the damper's resistance to the resistance of the system in which it is installed. Dickey and Coplen[2] utilized the resistance of the damper at its 30 percent open position in relation to the total system resistance to

[2] A Study of Damper Characteristics, by P. S. Dickey and H. L. Coplen (*ASME* Transactions, Vol. 64, 1942, pp. 137–154).

predict damper performance. In this investigation, the ratio of the wide open damper resistance to system resistance has been used as an index of the damper's ability to control air flow (the ratio A/B in Fig. 3). In either case, if the damper resistance constitutes a large part of the total system resistance, any change in damper position will alter the system resistance sufficiently to increase or decrease the air flow through the system.

Fig. 7 shows typical flow characteristics of parallel-leaf dampers. The left half of Table 2 lists the damper resistance as a percent of system resistance which must exist for each typical curve of Fig. 7.

Curve 1 of Fig. 7 represents a poor flow characteristic, since a great amount of damper movement is required to reduce the flow. In this case the open-damper resistance is one percent or less of the system resistance. (See Table 2). Consequently, the leaves must be moved a great number of degrees from the open position before sufficient resistance is added to the original system to reduce the flow. The damper controls 90 percent of the flow in the first 30 deg of movement from the closed position. It is desirable to control the flow uniformly over the full 90 deg of damper movement.

TABLE 2—RATIOS OF DAMPER RESISTANCE TO SYSTEM RESISTANCE WHICH APPLY TO FLOW CHARACTERISTICS IN FIGS. 7 AND 8

Fig. 7 Parallel-Leaf Dampers		Fig. 8 Opposed-Leaf Dampers	
Open Damper Resistance in Percent of System Resistance	Flow Characteristic	Open Damper Resistance in Percent of System Resistance	Flow Characteristic
0.5 — 1.0	1	0.25 — 0.50	1
1.0 — 1.5	2	0.50 — 0.75	2
1.5 — 2.5	3	0.75 — 1.50	3
2.5 — 3.5	4	1.5 — 2.5	4
3.5 — 5.5	5	2.5 — 5.5	5
5.5 — 9.0	6	5.5 — 13.50	6
9.0 — 15.0	7	13.5 — 25.5	7
15.0 — 20.0	8	25.5 — 37.5	8
20.0 — 30.0	9		
30.0 — 50.0	10		

Curve 8 represents a good flow characteristic since flow is nearly directly proportional to damper position. Several combinations of approach velocity and system resistance may result in the same ratio of damper resistance to system resistance and, consequently, the same flow characteristic. Table 3 lists several conditions of system resistance and approach velocity for each typical flow characteristic curve.

Fig. 8 shows typical flow characteristics of opposed-leaf dampers. The right half of Table 2 lists the damper resistance as a percent of system resistance which must exist for each typical curve of Fig. 8. Table 4 lists the approach velocity and system resistance conditions which pertain to each curve of Fig. 8.

The curves of Figs. 7 and 8 apply to dampers with an L/R ratio of 1.0 but may also be applied to other dampers having L/R ratios of 0.3 to 1.2 without introducing an appreciable error. The curves are strictly applicable only to systems in which

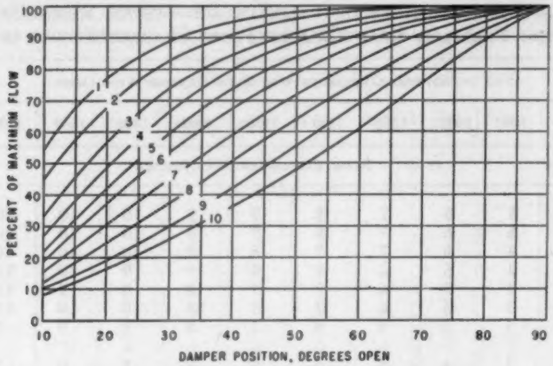


FIG. 7—PARALLEL-LEAF DAMPER FLOW CHARACTERISTICS

the static pressure at the entrance to the system is maintained constant from 0 to 100 percent flow. However, the curves present a general picture of the flow characteristics of parallel- and opposed-leaf dampers. Particularly, the curves illustrate that, for a given set of conditions, a better characteristic may be obtained with an opposed-leaf damper than with a parallel-leaf damper.

The following example will illustrate the steps which were necessary in determining each of the typical flow characteristic curves of Figs. 7 and 8.

Example: Assume that an opposed-leaf damper with an L/R ratio of 1.0 is installed in a system in which the approach velocity with the damper wide open is 2500 fpm. The resistance of the system including that of the open damper is 2.0 in. water. Determine the damper flow characteristic assuming that the fan will maintain a constant static pressure at its outlet. (The fan exit is considered to be the entrance of the system.)

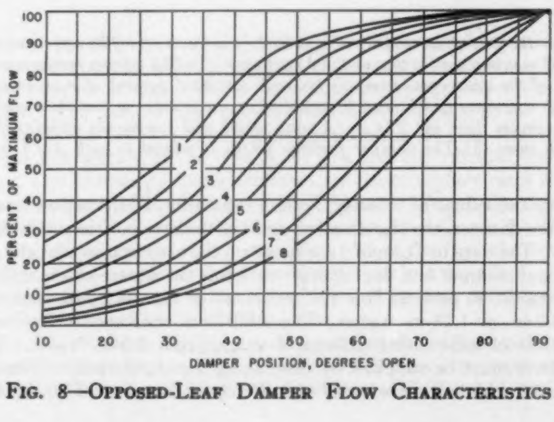


FIG. 8—OPPOSED-LEAF DAMPER FLOW CHARACTERISTICS

TABLE 3—TYPICAL CONDITIONS OF VELOCITY AND SYSTEM RESISTANCE WHICH APPLY TO THE PARALLEL LEAF DAMPER FLOW CHARACTERISTICS OF FIG. 7

Total[a] System Res., Inches Water	Approach Velocity fpm With Damper Wide Open										
	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000
	Flow Characteristic Number										
0.5	3	5	6	7	8	9	9	10	10	10	10
0.6	3	5	6	7	8	9	9	10	10	10	10
0.7	2	4	6	7	7	8	9	10	10	10	10
0.8	2	4	5	6	7	8	9	9	10	10	10
0.9	2	4	5	6	7	7	8	9	9	10	10
1.0	2	3	5	6	7	7	8	9	9	10	10
1.1	2	3	5	6	6	7	8	8	9	9	10
1.2	2	3	4	5	6	7	7	8	9	9	10
1.3	2	3	4	5	6	7	7	8	9	9	10
1.4	1	3	4	5	6	7	7	8	8	9	9
1.5	1	3	4	5	6	6	7	7	8	9	9
1.6	1	2	4	5	6	6	7	7	8	9	9
1.7	1	2	3	5	6	6	7	7	8	8	9
1.8	1	2	3	5	5	6	6	7	8	8	9
1.9	1	2	3	5	5	6	6	7	7	8	9
2.0	1	2	3	4	5	6	6	7	7	8	8
2.2	—	1	3	4	5	6	6	7	7	8	8
2.4	—	1	3	4	5	5	6	6	7	7	8
2.5	—	1	3	4	5	5	6	6	7	7	8
2.6	—	1	3	4	5	5	6	6	7	7	8
2.8	—	1	2	3	4	5	6	6	7	7	7
3.0	—	1	2	3	4	5	5	6	6	7	7

[a] Total system resistance is the sum of the resistances of the ductwork, the damper, and all other elements included in the system with the damper wide open and the maximum quantity of air flowing through the system.

To determine the flow characteristic it is necessary to know: (1) *The approach velocity for each percent of maximum flow that is to be tabulated*; (2) *The system resistance (including the resistance of the wide open damper) for each tabulated percent of maximum flow*; (3) *The additional resistance which must be supplied by the damper to maintain each tabulated percent of maximum flow*; (4) *The A_e/A ratio which will supply the additional resistance needed in each case*; (5) *The damper position which is related to each A_e/A ratio that is needed.*

Table 5 lists the values of velocity, system resistance, additional resistance to be supplied by the damper, the A_e/A ratios, and the damper positions related to these A_e/A ratios. The data in Column 3 are based on the assumption that the resistance of the wide open damper and duct system varies as the square of the volume of air flowing. Thus, at 80 percent flow the resistance of the duct and open damper is 0.64 times 2.0-in. or 1.28-in. water. The additional resistance values in Column 4 are the result of subtracting Column 3 values from 2.0-in. water. The additional resistance must be supplied by decreasing the A_e/A ratio. The resistance is in addition to that of the system and the open damper. The A_e/A ratio of

TABLE 4—TYPICAL CONDITIONS OF VELOCITY AND SYSTEM RESISTANCE WHICH APPLY TO THE OPPOSED LEAF DAMPER FLOW CHARACTERISTICS OF FIG. 8

Total[a] System Res., Inches Water	Approach Velocity fpm With Damper Wide Open										
	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000
	Flow Characteristic Number										
0.5	3	5	6	6	7	7	7	8	8	8	8
0.6	3	4	5	6	6	7	7	7	8	8	8
0.7	3	4	5	6	6	6	7	7	7	8	8
0.8	3	4	5	5	6	6	7	7	7	8	8
0.9	3	4	5	5	6	6	6	7	7	8	8
1.0	3	4	5	5	6	6	6	7	7	7	8
1.1	2	3	4	5	6	6	6	6	7	7	7
1.2	2	3	4	5	5	6	6	6	7	7	7
1.3	2	3	4	5	5	6	6	6	6	7	7
1.4	2	3	4	5	5	6	6	6	6	7	7
1.5	2	3	4	5	5	6	6	6	6	7	7
1.6	2	3	4	5	5	5	6	6	6	7	7
1.7	2	3	4	4	5	5	6	6	6	6	7
1.8	2	3	4	4	5	5	6	6	6	6	7
1.9	1	3	3	4	5	5	6	6	6	6	6
2.0	1	3	3	4	5	5	6	6	6	6	6
2.2	1	3	3	4	5	5	5	6	6	6	6
2.4	1	2	3	4	4	5	5	5	6	6	6
2.5	1	2	3	4	4	5	5	5	6	6	6
2.6	1	2	3	4	4	5	5	5	6	6	6
2.8	1	2	3	3	4	5	5	5	6	6	6
3.0	1	2	3	3	4	5	5	5	6	6	6

[a] Total system resistance is the sum of the resistances of the ductwork, the damper, and all other elements included in the system with the damper wide open and the maximum quantity of air flowing through the system.

0.36 necessary for 80 percent flow (2000 fpm) was obtained from Fig. 5 by proceeding vertically on the 2000 fpm line to the pressure loss line of 0.85 in. water which is 0.72 in. water above the resistance of the open damper (0.13 in. water) at 2000 fpm. The velocity and pressure loss lines intersect at an A_e/A ratio of 0.36 (see arrows on Fig. 5). The damper position of 69 deg was obtained from Fig. 6 at the intersection of the 0.36 A_e/A line and the 1.0 L/R line (see arrows on Fig. 6). The damper positions necessary for the other flow rates were obtained in the same manner. The flow rates in Column 1 plotted against the damper positions in Column 5 describe the flow characteristic. Curve 6 of Fig. 8 represents the flow characteristic of this example. All of the flow characteristics presented in Figs. 7 and 8 were developed by the method just described. The validity of the flow characteristic curves has been confirmed by tests conducted under conditions similar to those listed in Tables 3 and 4.

The exact flow characteristic for any damper-duct-fan system may be developed from the information given in Figs. 5 and 6. However, the representative flow characteristics of Figs. 7 and 8 and the conditions of Tables 2, 3, and 4 necessary

TABLE 5—TABULATED VALUES REQUIRED FOR EXAMPLE

1	2	3	4	5	6
FLOW IN PERCENT OF MAX. FLOW	APPROACH VELOCITY, FPM	TOTAL SYSTEM RESISTANCE INCLUDING OPEN DAMPER, INCHES WATER	ADDITIONAL RESISTANCE REQUIRED FROM DAMPER, INCHES WATER	A_0/A RATIO TO SUPPLY ADDITIONAL RESISTANCE	DAMPER POSITION CORRESPONDING TO A_0/A RATIO
100	2500	2.0	0	—	—
90	2250	1.62	0.38	0.44	77
80	2000	1.28	0.72	0.36	69
70	1750	0.98	1.02	0.29	63
60	1500	0.72	1.28	0.24	58
50	1250	0.50	1.50	0.20	53
40	1000	0.32	1.68	0.16	48
30	750	0.18	1.82	0.12	42
20	500	0.08	1.92	0.08	35
10	250	0.02	1.98	0.05	25

for these characteristics to exist eliminate the need for developing flow characteristics except for special cases which are not covered by these curves.

EXAMPLES OF THE USE OF THE FLOW CHARACTERISTIC CURVES

Example 1: *A parallel-leaf damper is installed in a branch duct. The velocity in the branch duct is* 1000 *fpm. The static pressure at the branch duct inlet is* 1.0 *in. water and is assumed to remain constant regardless of the flow in the branch duct. (In this case the system consists of the branch duct and any equipment such as coils, filters, or the damper included in the duct. The system resistance is equal to the static pressure of* 1.0 *in. water.)*

1. *What flow characteristic will result from these conditions?*

2. *If the parallel-leaf damper is replaced by an opposed-leaf damper, what flow characteristic will result?*

Solution:

1. Referring to Table 3, the velocity and system resistance conditions indicate that Curve 5 of Fig. 7 is the representative flow characteristic for a parallel-leaf damper in this system.

2. Referring to Table 4, the velocity and system resistance conditions indicate that Curve 5 of Fig. 8 is the representative flow characteristic for an opposed-leaf damper in this system. The opposed-leaf damper has a more desirable characteristic than does the parallel-leaf damper.

Example 2: *An opposed-leaf damper is installed in a* 24 *in.* x 24 *in. duct. With the damper wide open,* 16,000 *cfm of standard air flows through the system. The resistance of the system including the damper is* 4.0 *in. water. What volume of air will flow through the system if the damper position is changed to* 35 *deg?*

Solution: The flow of 16,000 cfm results in an approach velocity of 4000 fpm. These conditions of velocity and system resistance are not listed in Table 4. However, the flow characteristic may be determined from Table 2 and Fig. 8 if the ratio of the damper's resistance to the system resistance is established. Since the A_0/A ratio of the open damper is equal to 0.58, the resistance of the open damper in terms of approach velocity heads as given by Fig. 4 is 0.5 velocity heads. The velocity head for 4000 fpm is approximately 1.0 in. water. The damper resistance is then 0.5 in. water. The ratio of

damper resistance to system resistance is 0.5/4.0 or 12.5 percent. 12.5 percent is within the range of 5.5 to 13.5 percent listed for flow characteristic 6 in Table 2. Curve 6 of Fig. 8 indicates a flow of 20 percent of maximum for the 35 deg position. The flow rate for the 35 deg position is 20 percent of 16,000 cfm or 3200 cfm.

Conclusions

1. The flow of air as controlled by a damper is dependent on the damper's ability to change the resistance of the system as its position is changed.
2. The resistance of a damper at any position of its leaves is dependent on its equivalent contracted area ratio at that position.
3. The contracted area ratio for any given leaf position varies with the damper's leaf length to frame perimeter (L/R) ratio.
4. The pressure loss of an open damper of the types tested is equivalent to ½ of a velocity head referred to the approach velocity.
5. A flow characteristic for any damper-fan-duct system may be developed if the fan static pressure characteristic is known and data are available which relate damper resistance to leaf position and approach velocity.
6. If a constant static pressure is assumed at the system entrance for 0 to 100 percent flow, the flow characteristics of Figs. 7 and 8 may be applied to any conditions listed in Table 2 or Tables 3 and 4.
7. In most cases for the same conditions of approach velocity and system resistance the flow characteristic of an opposed-leaf damper is better than the flow characteristic of a parallel-leaf damper.

DISCUSSION

H. B. Nottage, Encino, Calif., (Written): 1. To what extent, and in what quantitative manner, can the results presented be applied to damper geometries and flow rates outside of the range of the test data?

2. What practical improvements in damper design can be set forth from the results?

3. What are the practical merits of introducing the *equivalent contracted area ratio* as opposed to expressing the net loss in velocity heads or in equivalent length of straight duct with stated friction factor?

4. Of the total net loss, what part is really a *sudden expansion* loss and what other constituent losses can be identified? Is the reported net loss in error because of the change in duct friction immediately following the damper where the duct velocity distribution would not be that in fully developed normal flow?

J. N. Livermore, Detroit, Mich., (Written): This is a practical and useful paper which should be widely and carefully read by designers of air handling systems. For many years the importance of proper selection of valves for the control of liquid and gas flow has been well understood and good data have been available. This has not been the case for the air flow control damper,—and particularly neglected has been the widely used multiple-leaf or louver damper on which the authors have here concentrated. It is my hope that this new information will find its place in The Guide without delay, and that its presence there be pointed out with appropriate emphasis.

The authors have probably purposely omitted one item of information which I think should be added. That is the rate of flow or leakage with the damper fully closed, which might be expected for each damper tested under various shut-off pressures. Figs. 7 and 8 show percent flows for 10 degree and wider damper openings. However, many curves in these figures seem to indicate that near zero flow would result at zero opening. This is scarcely believable for commercial dampers, particularly in cases of high pressure difference and for dampers of large leaf length to perimeter ratio. The control of systems operating on wide air temperature spreads has been known to fail

TABLE A

OPENING	DAMPER CLOSES ON 45°			AUTHORS' DATA		
	DAMPER POS., DEG.	$\frac{H_t}{VP}$	$\frac{A_e}{A}$	DAMPER POS., DEG.	$\frac{H_t}{VP}$	$\frac{A_e}{A}$
wide open	90	2.55	0.39	90	0.5	0.58
¼ open	78¾	3.6	0.35	67½	4.5	0.32
½ open	67½	6.9	0.28	45	50	0.13
¾ open	56¼	31	0.15	22½	570	0.041
closed	45	301	0.054	0		

because of damper leakage flow alone. Obviously, exact data are not necessary on this point, but the order of magnitude of shut-off leakage to be expected could be an important consideration when selecting a damper.

H. E. STRAUB, Waterloo, Ia., (WRITTEN): The authors have been very thorough in their presentation of a method to utilize basic damper information to develop damper-system flow characteristics. This procedure should be very valuable in the design of air-conditioning systems. The damper-system relationships indicate the careful considerations that are necessary for accurate and sensitive damper control in conventional systems and especially in high-pressure systems where the wide open damper loss may be only a small part of the system loss. It is evident from this paper that a damper alone cannot be called *linear* except when the connotation also includes the system characteristics.

Since the contracted area ratio is completely identified by the pressure ratio as given in Fig. 4, in Fig. 5 and 6 they could just as well have been shown as the primary quantity of pressure ratio. Thus the A_e/A ratios of 0.04, 0.10 and 0.20 for example could have been shown as an H_t/VP of 570, 78 and 15.5. As stated in the paper, the value of H_t/VP is practically constant for a given damper in a given leaf position. The primary information of H_t/VP obtained by a simple test then could be used directly in a revised Fig. 5 without the necessity of referring the pressure ratio to an area ratio.

The reference to Fig. 4 implies that the damper losses are due to an abrupt expansion, yet the discussion mentions free area repeatedly. Kratz and Kanzo showed in the University of Illinois *Bulletin* No. 342 that a variable border on the end of a duct followed the trend of the abrupt expansion curve. However, a damper breaks the air stream up into a multiple number of parallel air streams which reform into a single stream after passing through the damper. The only circumstances that provide enough space to approach an abrupt expansion before the single jet is reformed is when an opposed blade damper is nearly closed.

It appears that a grille would approach an abrupt expansion more nearly than a damper because it does not have the confining ductwork following the contractions. We have found that H_t/VP is generally related to the ratio of free area to duct area for grilles by $\frac{H_t}{VP} = \left(\frac{A_d}{A_f}\right)^2$. This relationship plots as a straight line on Fig. 4 with a slope of -2 and passes through the points $A_e/A = 1$, $H_t/VP = 1$. This curve shows a greater pressure loss than an abrupt expansion because a portion of the duct velocity head is lost and the actual contracted area is less than the free area. The curve also becomes asymptotic to the abrupt expansion curve at low area ratios. Other outlets, regardless of free area, such as ceiling outlets or baseboard outlets where the air stream makes a full 90 deg turn reaches a value of minimum loss which is similar to that of an elbow.

It appears that these concepts can be applied to the damper losses, a turning loss plus expansion with parallel leaf dampers and expansion with the opposed leaf. It

would be of interest to know if these data, based on free area, followed the curve for an abrupt expansion or if they followed more nearly a straight line curve. Also it would appear that the width of the leaf would have an effect. That is a narrow leaf would have a lower loss than a wide leaf because of the smaller space for expansion. Would the authors have any information on this?

I would like to re-emphasize that these curves apply only to the types of dampers covered in these studies. We have tested an opposed leaf damper with an L/R value of 1.5 which closes on a 45 deg angle and found the results shown in Table A.

Comparison of these data shows that except at the wide open position the opposed leaf damper which closes on a 45 deg angle has a higher contracted area ratio or lower pressure loss for a given opening than an opposed leaf damper which closes flat. The damper which closes on a 45 deg angle would then have a flow characteristic quite different from that shown by the authors for an opposed leaf damper. These results could be predicted by comparing the free area of these dampers at comparable openings.

Although free area may be one of the principal factors affecting the pressure loss and may be a means of predicting trends in damper design, the authors have rightly indicated the difficulty in measuring these areas and avoided using them in the application procedure. For the practical application they have pointed out that the information needed on dampers is the relationship of the pressure ratio to leaf position. If I don't, somebody else will add to this the sound characteristics which tests have indicated may also be related to the pressure ratio.

S. F. GILMAN and O. W. CLAUSEN*, Syracuse, N. Y., (WRITTEN): In this paper, considerable importance is placed on the A_e/A ratio, designated by the authors as the *equivalent contracted area ratio*. It is equal to the inverse of A_2/A_1 which appears in Equation 3 and the unnumbered equation following it; for ease of reference, we will call the latter Equation 4. (The reason for the sudden change in subscripts in the middle of the paper is not given.) We question the need and usefulness of this ratio; moreover, it will be demonstrated that it causes misleading impressions of the data presented.

At the beginning of the section *Resistance of Dampers to Air Flow*, it is pointed out that the free area of the damper is important but very difficult to measure. Supposedly to avoid this difficulty, the pressure losses are expressed in terms of the A_e/A ratio which appears in Equations 3 and 4 as A_1/A_2. However, since the left side of Equation 4 is the pressure loss divided by the velocity head referred to the duct area *upstream* (or *downstream*) of the damper, the problem of evaluating the free area within the damper blades never appears. Only if the authors had chosen to use the velocity head corresponding to the velocity within the damper as reference would the problem have arisen. Since they did not, there is no need for introducing the A_e/A ratio.

Now in several ASHAE papers the right hand side of Equation 4 has been denoted as λ and termed the *loss coefficient*. This method of expressing results is particularly convenient because the pressure loss in inches of water is simply obtained by multiplying the velocity pressure by λ. This procedure is also used in the ASHAE GUIDE; for example, Tables 2 and 3 in Chapter 31 lists loss coefficients for a multiplicity of duct fittings. What, then, are the advantages of introducing an equivalent abrupt expansion concept so that the right hand side of Equation 4 becomes $(A_2/A_1 - 1)^2$ instead of simply λ? We see none whatsoever, but we do see a major disadvantage. By using $(A_2/A_1 - 1)^2$, or what is the same thing, $(A/A_e - 1)^2$, and expressing the results in terms of A_e/A, the correlation of the data appears better than it actually is.

Referring to Fig. 6, the data appear to correlate well for the 90 deg settings (wide open) and scatter considerably at the small angles. However, it is not A_e/A that has been determined experimentally. What has been measured is H_t and V_2 in Equation 4 so that the experimentally determined value is $(A/A_e - 1)^2$; i.e., the right hand side of

* Research Engineer, Carrier Corporation.

Equation 4. Solving this with A_e/A equal to a high of 0.61 and low of 0.51 from the 90 deg opposed leaf damper data in Fig. 6, we obtain 0.41 and 0.92. Hence, at the same approach velocity, the measured pressure losses vary by a factor of 2.24. We conclude that the experimental data actually scatters widely for the 90 deg setting. For the 10 deg setting of the opposed leaf dampers, at an L/R of about 0.72, the high is approximately 0.039 and the low 0.025. The right hand side of Equation 4 then becomes 610 and 1520, respectively. Hence the loss coefficients differ by a factor of 2.5, only slightly greater than for the 90 deg setting. Using the Fig. 6 results, more than 30 corresponding loss coefficients were calculated and plotted against L/R, with the conclusion that the experimental data show a very wide scatter.

The A_e/A ratio depresses the scatter of the experimental data and hence gives misleading impressions. Moreover, it has no physical significance. It is uniquely related to $H_t/(V_2^2/2g)$ so that the ordinates of Fig. 6 could just as well be $H_t/(V_2^2/2g)$ or, what is the same thing, λ. Hence the ratio A_e/A is only an additional intermediate parameter that is not needed. Consequently, we strongly urge the A_e/A concept be abandoned and an attempt be made to correlate the data on a loss coefficient basis.

The Fig. 6 scatter raises doubts that A_e/A can actually be correlated with L/R. The paper defines L/R as *the sum of the lengths of all of the leaves divided by the perimeter of the duct.* The physical significance of this parameter can be analyzed by expressing it in equation form, which is, approximately:

$$\frac{L}{R} = \frac{n\,W}{2(H + W)}$$

where

n = number of leaves
W = leaf length and also, to a close approximation, the duct width
H = duct height

From this it is evident that the number of leaves, n, strongly influences the value of L/R; in our opinion, the number of leaves is given far more influence than it deserves. If the leaf length, W, is increased, the influence is not so great because W appears both in the numerator and denominator. The influence of H depends on W. If W is small, the influence of H can be great. However, if W is large, a considerable change in H can have little effect on L/R. Hence L/R is a function of 3 variables that are interrelated in a complex manner. It appears the probability of good correlation of A_e/A with L/R is very slight.

This is borne out by a study of the middle 3 dampers in the third group down in Table 1 for the 20 deg setting. Here the major variable is the number of leaves. Yet instead of decreasing with L/R, the resultant values of A_e/A are very erratic, being 0.07, 0.038 and 0.056. It is therefore suggested that the authors attempt to find a less complex parameter for correlation purposes, such as W/H. Would the authors care to comment on the reasoning leading to the selection of L/R for correlating the data?

It appears that a considerable amount of data has been collected, but the data are influenced by many variables which have not been isolated. We suggest that a more fundamental approach to the problem be made by measuring and analyzing the velocity and pressure patterns within and adjacent to the dampers. The effect of a certain portion of the flow bypassing the dampers at other than the 90 deg setting may be significant and should be determined. By systematic variation of variables and comprehensive internal flow measurements, significant information should be obtained.

The authors are commended for attacking such a difficult, multivariable problem and we sincerely hope they can carry it forward to a fruitful conclusion.

LEONARD PHILLIPS, Hartford, Conn. (WRITTEN): I sincerely thank the authors for this paper. It is an interesting example of one type of good paper. It takes a subject, and while disclosing no radical departure, or new development, points out how a good engineering approach is often overlooked and proceeds to provide a method, data and examples for a rational application of sound engineering practice.

Only too often, both custom-made and proprietary type dampers are applied by merely selecting a size to fit some convenient area where the air is to be restricted, such as the duct area, with no consideration of the damper itself.

Particularly with respect to dampers connected to control devices, the connection between damper position, damper resistance and system resistance is important and often overlooked.

I have three questions to ask the authors.

First: In leaf dampers using gaskets to effect tight closure between the leaves only, would there be any change in the application of the L/R ratio as given?

Second: The simple leaf dampers were tested up to 5½-in. water and by inference and use of the data given, can be applied to control air up to that pressure. Although it has no connection with the facts presented, I would appreciate a word on the question of how these dampers sounded at the higher flow rates and higher static pressure resistances?

Third: Can the information presented be used for other dampers, for example those similar in type, but of streamline design, in which the area change is less abrupt and the static pressure regain much higher?

I again wish to thank the authors and their sponsors for such valuable information which can be used by such a large cross section of our profession.

A. V. HARRINGTON, Boston, Mass.: We are currently interested in an installation presently being constructed using a dual-duct low pressure system. There are certain areas that must be supplied with a constant volume of air and the design includes the combination of dampers on a common shaft. There are one each and a cold and hot duct goes into a common duct which supplies the air to the space.

My question is the following one. If a pair of multiple leaf dampers were used in a dual-duct system, (that is, a similar unit in each duct and connected on a common shaft), what degree of regular air flow could be expected in the common outlet, assuming that it is provided with necessary controls designed to maintain equal pressure in both the hot and cold duct?

H. E. ZIEL, Detroit, Mich.: I would like to inquire if any data were obtained relative to the increase in the air stream sound level for various damper positions.

The conventional method of low pressure air supply in an office area consists of a distribution system where the duct work is installed above an acoustic ceiling with branches extended to ceiling outlets.

Occasionally in large office areas, a conference room or similar enclosure is installed where separate damper controlled branches are installed due to the intermittent use of the conference room which may increase the noise level in this enclosed room, especially when the damper is in the nearly closed position.

However, it has been pointed out by some acoustical engineers that a continuous added noise not too high in level, can actually be beneficial. Should the noise from an air handling system raise the sound threshold level it may mask out certain intermittent sounds which may be otherwise objectionable.

J. H. BURGES, New York, N. Y.: I would like to confine my point to the use of contracted area ratios (A_e/A) in Figs. 5 and 6.

It strikes me, from a practical point of view, that the effect of these ratios will vary for each manufacturer's type of damper. The shape and length of the edgehook and the possible use of side stops would cause variations. In other words, the manufacturer will have to provide correction factor information to apply to the authors' charts.

If the manufacturer has information to provide, isn't it better to relate cfm per square foot leakage (all velocities) and pressure drop directly to the blade angle instead of going through this extra (and sometimes impractical) process in finding A_e/A? This is already done by some manufacturers. The information is available in this form from some manufacturers. The extra steps, it seems to me, are unnecessary.

AUTHORS' CLOSURE (Mr. Brown): The comments which have been presented indicate the limitations of the pressure loss data. Several comments questioned the need for the

introduction of the equivalent contracted area ratio concept. The total pressure losses could as well have been expressed directly in terms of velocity heads. In that case the losses range from ½ to over 2000 velocity heads for the 90 deg and 10 deg positions, respectively. In terms of equivalent length of duct the losses range from 20 ft to over 10,000 ft for the 90 deg and 10 deg positions, respectively. The contracted area ratio concept was used as a means of relating the pressure losses to those which occur in abrupt expansion.

Mr. Gilman and Mr. Clausen have shown that the L/R ratio is not the correct parameter for correlating the pressure loss data because of the effect that the number of leaves has on this ratio. Their suggestion of attempting a correlation of data based on the aspect ratio of the dampers will be pursued.

Mr. Nottage and Mr. Straub have inquired as to the possibility of separating the pressure losses into abrupt expansion losses and losses due to friction. In order to do this further pressure loss measurements close to the damper would be required. A breakdown of the total loss into components would be helpful to one concerned with damper design.

Mr. Straub has also brought out another point by comparing the damper which closes at an angle of 45 deg with the damper which must operate through a full 90 deg cycle. That is, the flow characteristic curves presented in this paper should be applied only to the particular type of dampers described. This does not mean that the flow characteristics data are restricted to one manufacturer's dampers. Dampers of three manufacturers were included in the study. The dampers were quite similar as far as leaf construction and operation of the leaves were concerned. The dampers were also similar as far as pressure losses and flow characteristics were concerned. Dampers with leaves which decidedly differ in design from those shown in Fig. 1 would be expected to have somewhat different flow characteristics.

Mr. Livermore asked about leakage. Leakage was studied as a separate phase of the investigation and expressed as a function of leaf length and pressure difference across the closed damper. It was found that leaves longer than 2 ft were subject to considerable bending at the closed position with pressure differences greater than 2 in. of water. The bending caused a considerable flow area to be opened up between the damper frame and the leaf edges parallel to the leaf axis. The flow characteristic curves were not restricted to any one leaf length, and, therefore, the effect of leakage at the 0 deg position is not shown. If leakage through the closed damper is of importance in the design of a particular system then the dampers selected for that system should have some type of treatment which will restrict flow between the leaves and damper frame.

Mr. Ziel and Mr. Phillips asked about sound-level measurements. No study was made of sound level or changes in sound level with a change in damper position. When a general agreement has been reached as to how sound-level measurements should be presented such a study would provide useful information.

Mr. Harrington asked about the possibility of maintaining a constant volume of air at a single outlet by means of interconnecting dampers in hot and cold air ducts supplying air to the outlet. The total volume supplied by the two ducts would be constant if the flow characteristics of both dampers were linear (similar to curve 9, Fig. 7). However, if one damper had a characteristic similar to curve 1 of Fig. 7 and the other damper had a characteristic similar to curve 10, Fig. 7, there would be no way of maintaining a constant air-flow rate at the outlet through a full 90 deg cycle of operation of the interconnected dampers.

The authors wish to thank everyone for their comments. The discussions indicate that the paper's greatest value has been to show that the damper must be considered in the design of air-conditioning systems. The discussions also show that further work should be concerned with data that would be of use to the damper designer.

1638

No. 1638

FAN NOISE VARIATION WITH CHANGING FAN OPERATION

By R. D. MADISON* AND J. B. GRAHAM**, BUFFALO, N. Y.

EVALUATION of ventilation system noise in occupied spaces has assumed greater importance in the past few years, and the engineering factors in this evaluation have been under study by many people. The ASHAE has sponsored work at its own laboratory in Cleveland, which resulted in the presentation of a paper[1] at the 1957 Semi-Annual Meeting at Murray Bay. Progress has been made toward establishing a new sound code for the heating and ventilating industry that will be of material assistance to engineers in designing new systems or in improving the acoustic characteristics of existing systems. The background material on this subject was extremely well covered at the summer ASHAE Meeting, at which time a number of papers were presented and a Symposium was held[2]. The subject material covered was broad in scope, but all material was aimed at evaluation of system noise. It is generally agreed that one of the most important factors in determining system noise is the evaluation of fan noise.

The problem of fan noise has been approached by an evaluation of methods of determining the sound power of a fan, techniques of measurement, proposed standards and application of standards. Previous papers on fan noise have been concerned primarily with techniques of determining sound power, with proposed codes and standards, with noise criteria and with applications of fan noise data to ventilating systems. The purpose of this paper is to discuss the fan as a source of noise and to describe the changes in fan sound power as a function of capacity, pressure, size and speed.

FAN TYPES

To investigate the sound-power characteristics of fans, several types of centrifugal, axial and propeller fans were studied. Fan types were so selected as to permit use of geometrically similar fans to provide wide variation of parameters

* Consulting Engineer, Buffalo Forge Co. Member of ASHAE.
** Director of Research, Buffalo Forge Co. Member of ASHAE.
[1] Exponent numerals refer to References.
Presented at the 64th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Pittsburgh, January 1958.

FIG. 1—FAN TYPE INCLUDED IN STUDY

FIG. 2—ANOTHER TYPE OF FAN STUDIED

FIG. 3—A THIRD TYPE OF FAN STUDIED

so that excessive extrapolation would not be necessary. Some of the various fans used in this study are shown in Figs. 1, 2 and 3.

ROOM

A room 66 ft long x 44 ft wide x 16 ft high (46,500 cu ft) was used for the sound-power investigation of these fans. The room was considered to approximate the requirements for a reverberant space. It was, in general, acceptable under some of the proposed test codes. Calibration of the room was made by using the same noise generator used in the evaluation studies at the ASHAE Laboratory. Overall sound-power level readings and octave-band power-level values for the noise generator were the same as used by the Research Laboratory. A survey of the room, using a rectangular co-ordinate system, showed that it would be advisable to use the average of 4 or 5 readings to obtain good values for evaluation. Variations and averages of sound-pressure level for this room, using the sound generator as a noise source, are shown in Fig. 4. In the comparisons described herein the average values from this curve were used. The room construction was of concrete block walls, concrete floor and steel pan construction ceiling.

INSTRUMENTATION AND TECHNIQUES OF MEASUREMENTS

Sound-pressure-level measurements were made using a Type 759-B sound-level meter coupled to a Type 420-A octave-band analyser for frequency analysis.

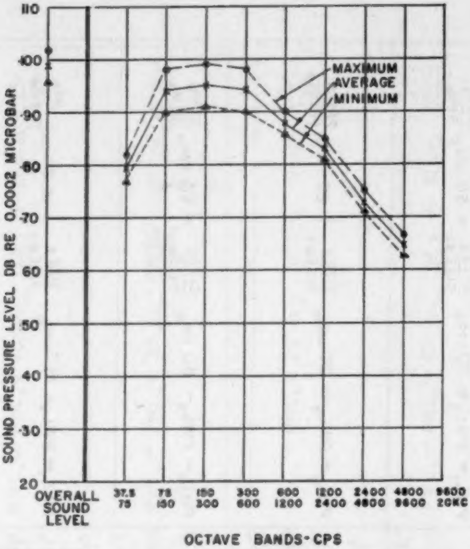


FIG. 4—SOUND-PRESSURE-LEVEL VALUES FOR REVERBERANT SPACE USING SOUND GENERATOR AS NOISE SOURCE

Measurements were made on the centrifugal and axial fans by setting up standard *NAFM* test duct arrangements. The fan was located in the reverberant room with the discharge duct projecting through the wall into an adjacent room and no duct on the fan inlet. Readings of sound-pressure level were taken in the reverberant field around the fan, with no reading taken closer than 10 ft. Power-level values were then calculated, using the formulae shown in the Appendix. It is recognized that sound-pressure measurements taken in this manner are sound pressures that result from inlet noise and radiated sound from the fan housing. Of more importance for most noise evaluations is the sound power radiated by the fan into the discharge duct. At the time of these tests there was not a completely acceptable method of determining sound power in the discharge duct. However, it is felt that the techniques used in this study provide accurate data to establish the sound-power relationships for changes in fan capacity, pressure, size and speed. When an acceptable method is established, further work will be done to determine the necessary correlation between the present data and the new data.

Directivity was not a factor in this study, since the reverberant room technique does not lend itself to a study of this kind.

FAN LAWS

The performance curve of a centrifugal fan is shown in Fig. 5. Typically, these curves are drawn for a given size of fan of a specific type and operating at a given

Table 1—Tabular Representation of Fan Laws and of Sound Laws for Fans

Law No.	Fan Laws: Depend. Variable		Basic Data		Independent Variables	Density Corr.	Sound Laws for Fans
1	CFM_S	=	CFM_B	×	$\left(\frac{SIZE_S}{SIZE_B}\right)^3 \times \left(\frac{RPM_S}{RPM_B}\right)$	× (1)	$PWL_S = PWL_B + 70 \log_{10} \frac{SIZE_S}{SIZE_B} + 50 \log_{10} \frac{RPM_S}{RPM_B}$
	$PRESS_S$	=	$PRESS_B$	×	$\left(\frac{SIZE_S}{SIZE_B}\right)^2 \times \left(\frac{RPM_S}{RPM_B}\right)^2$	$\times \left(\frac{\rho_S}{\rho_B}\right)$	
	HP_S	=	HP_B	×	$\left(\frac{SIZE_S}{SIZE_B}\right)^5 \times \left(\frac{RPM_S}{RPM_B}\right)^3$	$\times \left(\frac{\rho_S}{\rho_B}\right)$	
2	CFM_S	=	CFM_B	×	$\left(\frac{SIZE_S}{SIZE_B}\right)^2 \times \left(\frac{PRESS_S}{PRESS_B}\right)^{1/2}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/2}$	$PWL_S = PWL_B + 20 \log_{10} \frac{SIZE_S}{SIZE_B} + 25 \log_{10} \frac{PRESS_S}{PRESS_B}$
	RPM_S	=	RPM_B	×	$\left(\frac{SIZE_B}{SIZE_S}\right) \times \left(\frac{PRESS_S}{PRESS_B}\right)^{1/2}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/2}$	
	HP_S	=	HP_B	×	$\left(\frac{SIZE_S}{SIZE_B}\right)^2 \times \left(\frac{PRESS_S}{PRESS_B}\right)^{3/2}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/2}$	
3	RPM_S	=	RPM_B	×	$\left(\frac{SIZE_B}{SIZE_S}\right)^3 \times \left(\frac{CFM_S}{CFM_B}\right)$	× (1)	$PWL_S = PWL_B - 80 \log_{10} \frac{SIZE_S}{SIZE_B} + 50 \log_{10} \frac{CFM_S}{CFM_B}$
	$PRESS_S$	=	$PRESS_B$	×	$\left(\frac{SIZE_B}{SIZE_S}\right)^4 \times \left(\frac{CFM_S}{CFM_B}\right)^2$	$\times \left(\frac{\rho_S}{\rho_B}\right)$	
	HP_S	=	HP_B	×	$\left(\frac{SIZE_B}{SIZE_S}\right)^4 \times \left(\frac{CFM_S}{CFM_B}\right)^3$	$\times \left(\frac{\rho_S}{\rho_B}\right)$	
4	CFM_S	=	CFM_B	×	$\left(\frac{SIZE_S}{SIZE_B}\right)^{4/3} \times \left(\frac{HP_S}{HP_B}\right)^{1/3}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/3}$	$PWL_S = PWL_B - 13.3 \log_{10} \frac{SIZE_S}{SIZE_B} + 16.6 \log_{10} \frac{HP_S}{HP_B}$
	$PRESS_S$	=	$PRESS_B$	×	$\left(\frac{SIZE_B}{SIZE_S}\right)^{4/3} \times \left(\frac{HP_S}{HP_B}\right)^{2/3}$	$\times \left(\frac{\rho_S}{\rho_B}\right)^{1/3}$	
	RPM_S	=	RPM_B	×	$\left(\frac{SIZE_B}{SIZE_S}\right)^{5/3} \times \left(\frac{HP_S}{HP_B}\right)^{1/3}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/3}$	
5	$SIZE_S$	=	$SIZE_B$	×	$\left(\frac{CFM_S}{CFM_B}\right)^{1/2} \times \left(\frac{PRESS_B}{PRESS_S}\right)^{1/4}$	$\times \left(\frac{\rho_S}{\rho_B}\right)^{1/4}$	$PWL_S = PWL_B + 10 \log_{10} \frac{CFM_S}{CFM_B} + 20 \log_{10} \frac{PRESS_S}{PRESS_B}$
	RPM_S	=	RPM_B	×	$\left(\frac{CFM_B}{CFM_S}\right)^{1/2} \times \left(\frac{PRESS_S}{PRESS_B}\right)^{3/4}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{3/4}$	
	HP_S	=	HP_B	×	$\left(\frac{CFM_S}{CFM_B}\right) \times \left(\frac{PRESS_S}{PRESS_B}\right)$	× (1)	

SUBSCRIPT "B" INDICATES VALUES FROM BASE CURVE OR CATALOG—"S" INDICATES VALUES FOR FAN SELECTION

TABLE 1 (Continued)—TABULAR REPRESENTATION OF FAN LAWS AND OF SOUND LAWS FOR FANS

LAW NO.	FAN LAWS: DEPEND VARIABLE	BASIC DATA	INDEPENDENT VARIABLES	DENSITY CORR.	SOUND LAWS FOR FANS
6	$SIZE_S =$	$SIZE_B$	$\times \left(\frac{CFM_S}{CFM_B}\right)^{1/3} \times \left(\frac{RPM_B}{RPM_S}\right)^{1/3}$	$\times (1)$	$PWL_S = PWL_B + 23.3 \log_{10} \frac{CFM_S}{CFM_B} + 26.6 \log_{10} \frac{RPM_S}{RPM_B}$
	$PRESS_S =$	$PRESS_B$	$\times \left(\frac{CFM_S}{CFM_B}\right)^{2/3} \times \left(\frac{RPM_S}{RPM_B}\right)^{4/3}$	$\times \left(\frac{\rho_S}{\rho_B}\right)$	
	$HP_S =$	HP_B	$\times \left(\frac{CFM_S}{CFM_B}\right)^{5/3} \times \left(\frac{RPM_S}{RPM_B}\right)^{4/3}$	$\times \left(\frac{\rho_S}{\rho_B}\right)$	
7	$SIZE_S =$	$SIZE_B$	$\times \left(\frac{PRESS_S}{PRESS_B}\right)^{1/2} \times \left(\frac{RPM_B}{RPM_S}\right)$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/2}$	$PWL_S = PWL_B + 35 \log_{10} \frac{PRESS_S}{PRESS_B} - 20 \log_{10} \frac{RPM_S}{RPM_B}$
	$CFM_S =$	CFM_B	$\times \left(\frac{PRESS_S}{PRESS_B}\right)^{3/2} \times \left(\frac{RPM_B}{RPM_S}\right)^{2}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{3/2}$	
	$HP_S =$	HP_B	$\times \left(\frac{PRESS_S}{PRESS_B}\right)^{5/2} \times \left(\frac{RPM_B}{RPM_S}\right)^{2}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{3/2}$	
8	$SIZE_S =$	$SIZE_B$	$\times \left(\frac{HP_B}{HP_S}\right)^{1/4} \times \left(\frac{CFM_S}{CFM_B}\right)^{3/4}$	$\times \left(\frac{\rho_S}{\rho_B}\right)^{1/4}$	$PWL_S = PWL_B + 20 \log_{10} \frac{HP_S}{HP_B} - 10 \log_{10} \frac{CFM_S}{CFM_B}$
	$RPM_S =$	RPM_B	$\times \left(\frac{HP_S}{HP_B}\right)^{3/4} \times \left(\frac{CFM_B}{CFM_S}\right)^{5/4}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{3/4}$	
	$PRESS_S =$	$PRESS_B$	$\times \left(\frac{HP_S}{HP_B}\right) \times \left(\frac{CFM_B}{CFM_S}\right)$	$\times (1)$	
9	$SIZE_S =$	$SIZE_B$	$\times \left(\frac{HP_S}{HP_B}\right)^{1/2} \times \left(\frac{PRESS_B}{PRESS_S}\right)^{3/4}$	$\times \left(\frac{\rho_S}{\rho_B}\right)^{1/4}$	$PWL_S = PWL_B + 10 \log_{10} \frac{HP_S}{HP_B} + 10 \log_{10} \frac{PRESS_S}{PRESS_B}$
	$RPM_S =$	RPM_B	$\times \left(\frac{HP_B}{HP_S}\right)^{1/2} \times \left(\frac{PRESS_S}{PRESS_B}\right)^{5/4}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{3/4}$	
	$CFM_S =$	CFM_B	$\times \left(\frac{HP_S}{HP_B}\right) \times \left(\frac{PRESS_B}{PRESS_S}\right)$	$\times (1)$	
10	$SIZE_S =$	$SIZE_B$	$\times \left(\frac{HP_S}{HP_B}\right)^{1/5} \times \left(\frac{RPM_B}{RPM_S}\right)^{3/5}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{1/5}$	$PWL_S = PWL_B + 14 \log_{10} \frac{HP_S}{HP_B} + 8 \log_{10} \frac{RPM_S}{RPM_B}$
	$CFM_S =$	CFM_B	$\times \left(\frac{HP_S}{HP_B}\right)^{3/5} \times \left(\frac{RPM_B}{RPM_S}\right)^{4/5}$	$\times \left(\frac{\rho_B}{\rho_S}\right)^{3/5}$	
	$PRESS_S =$	$PRESS_B$	$\times \left(\frac{HP_S}{HP_B}\right)^{2/5} \times \left(\frac{RPM_S}{RPM_B}\right)^{4/5}$	$\times \left(\frac{\rho_S}{\rho_B}\right)^{3/5}$	

SUBSCRIPT "B" INDICATES VALUES FROM BASE CURVE OR CATALOG— "S" INDICATES VALUES FOR FAN SELECTION

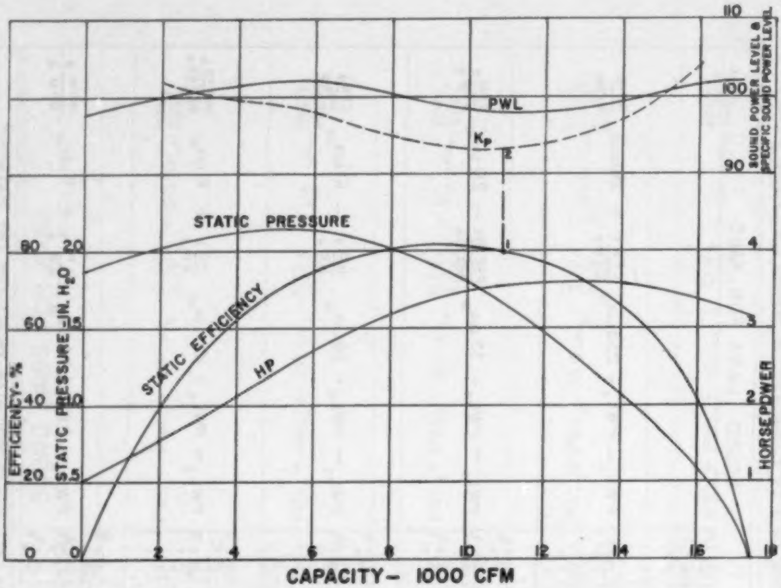


FIG. 5—PERFORMANCE CURVE OF A TYPICAL CENTRIFUGAL FAN

speed. Performance characteristics of the same type of fan, with size and speed as variables, may be determined by use of the fan laws. For reference, these laws are shown in Table 1. A knowledge of the operation of these fan laws is essential to an understanding of the Sound Laws for fans described later. Fan law calculations apply only to equivalent points of rating on a set of fan curves. This is illustrated in Fig. 6 where a *base curve* is shown with a *computed curve* and the equivalent points of rating "A" and "B" shown on each curve. This illustrates an example of a change in fan speed and is an application of Fan Laws 1, 2, 3 or 4.

It should be noted that point "A" is at 75 percent of free delivery capacity and 67 percent of shutoff static pressure and that the equivalent point of rating "B" on the lower speed curve is also at 75 percent of free delivery capacity and 67 percent of shutoff static pressure. Sample calculations are included in the Appendix.

In a similar manner, Fig. 7 illustrates the application of Fan Laws 2, 5, 7 or 9 and represents equivalent points of rating at constant pressure. In this case the performance curves represent different sizes of geometrically similar fans operating at different speeds. Fig. 8 illustrates the application of Fan Laws 3, 5, 6 and 8, and represents equivalent points of rating at constant capacity. Again, these performance curves represent different sizes of geometrically similar fans operating at different speeds.

It must be understood that there is no way to calculate the conditions at some random Point "C" (Figs. 6, 7, 8) on the computed curve by starting at Point "A" on the base curve. To compute conditions of Point "C," calculations must be

based on the equivalent point of rating on the base curve. These explanations illustrate several of the most useful fan law calculations but do not cover all possible combinations. A more complete explanation is contained in Reference 4.

Instead of using a particular performance curve as shown in Fig. 1 for fan calculations, it is often convenient to define the characteristics of a type of fan by a single curve. One way to do this is to make use of the specific speed of the fan. Specific speed in fan work is based on static pressure and is the revolutions per minute at which a fan of that type would operate to furnish 1 cfm at 1 in. static pressure.

$$N_s = \text{rpm} \times \text{cfm}^{1/2}/P_s^{3/4} \quad (1)$$

where

N_s = specific speed.
rpm = revolutions per minute of fan.
cfm = cubic feet per minute.
P_s = static pressure of fan, inches of water.

This will be recognized as a special application of Fan Law 5. Specific speeds are used to establish equivalent points of rating for geometrically similar fans of different size and operating at different speeds.

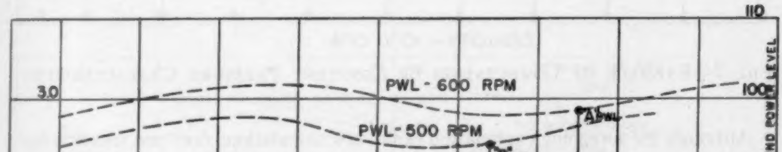


FIG. 6—EXAMPLE OF COMPUTATION OF A CHANGE OF SPEED OF THE TYPICAL FAN OF FIG. 5

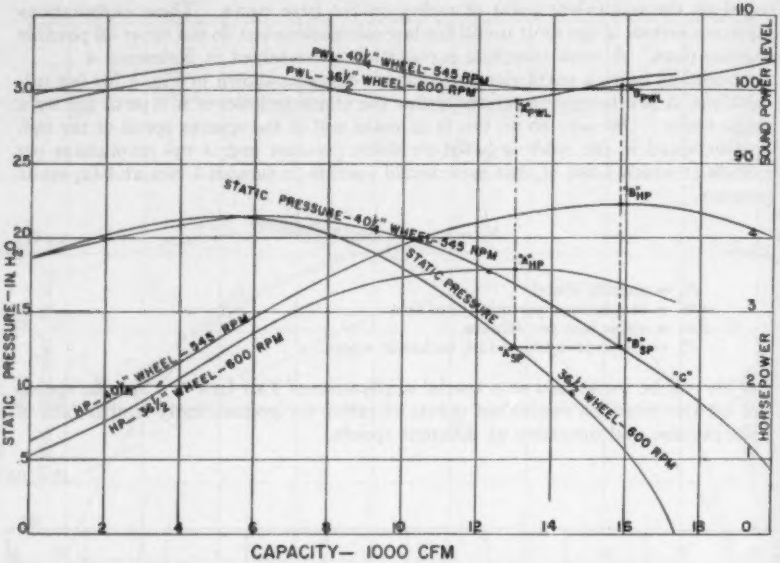


FIG. 7—EXAMPLE OF COMPUTATION OF CONSTANT PRESSURE CHARACTERISTICS

Although the foregoing explanation of fan law calculations does not mention fan sound power specifically, an understanding of these laws is necessary for the following discussion of sound power, since a definite relationship can be established between the fan laws and a parallel set of sound laws for fans.

FAN NOISE

The noise generated by a fan falls into the two general classifications of rotational noise and vortex noise.[3, 4] Rotational noise is created by a blade as it passes a given point and gives rise to a fundamental tone commonly known as the blade frequency.

$$f_B = \text{rpm} \times B/60 \qquad (2)$$

where

f_B = blade frequency, cycles per second.
rpm = revolutions per minute of fan.
B = number of blades on fan.

Harmonics of the fundamental tone will be present, but the fundamental tone will be dominant in the sound spectrum as shown by the typical octave-band analysis. The intensity of the blade noise is determined by the shape of the air impulse. Vortex noise is a broad-band noise in the fan due to turbulence or unstable flow in the fan. If the flow were laminar throughout, little vortex noise would be generated. However, there is usually some separation that generates

eddy flow. If the point of flow separation is variable, the eddy flow fluctuates rapidly and creates considerable noise. Even without this fluctuation, however, the eddy flow is a source of broad-band noise. Some of the vortex noise is contributed by Karman vortex trails shed from the trailing edges of the blades, and these vortices, since they are random in character, are likewise a source of broad-band noise. It is obvious from the foregoing that the lowest sound power will occur near the point of maximum efficiency, since this is the point of optimum flow conditions. Also from the foregoing it may be seen that the quantity of air flowing through the wheel and the pressure built up in the wheel will have important effects on the sound power generated by the fan. Furthermore, it is important to consider the size of the fan and the speed at which it operates to produce the given conditions when making calculations of sound power.

An organized approach to this problem is possible by making sound-power-level calculations on the same basis as other fan calculations and by restricting the calculations to equivalent points of rating.

As with most problems of this kind, it is impossible to vary only one of the parameters, while holding all others constant, and still maintain the stated restriction of equivalent points of rating. However, it is possible to investigate the effects of one variable by selecting a particular fan law, and while holding one of the independent variables constant, make changes in the other. This change of one independent variable will produce an equivalent point of rating with new values of the dependent variables; and also a sound-power level for the new point of rating.

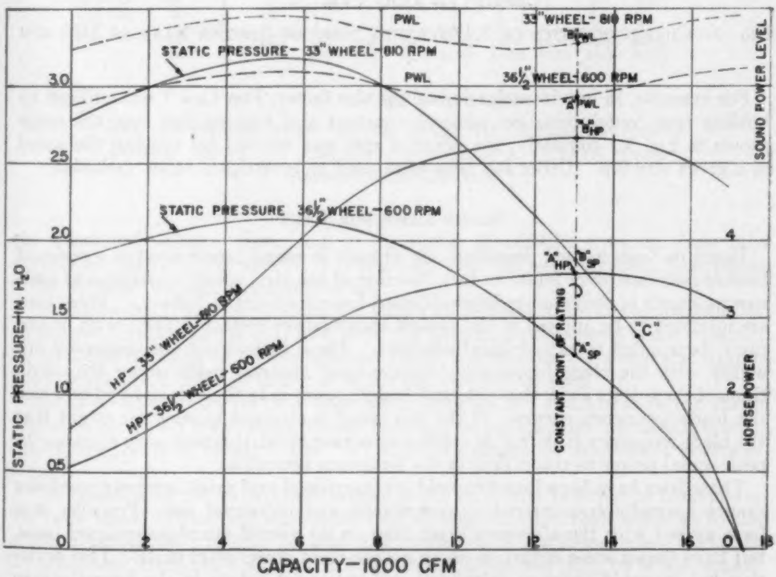


FIG. 8—EXAMPLE OF COMPUTATION OF CHARACTERISTICS AT CONSTANT CAPACITY

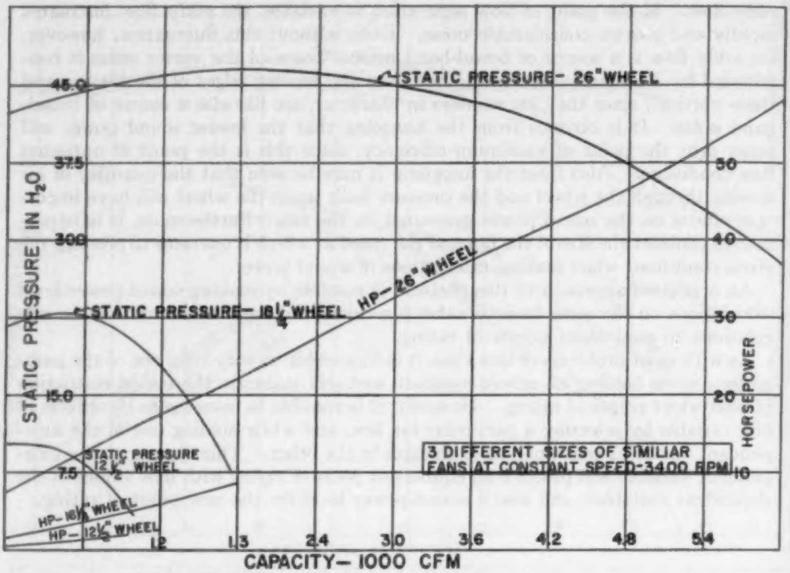


Fig. 9—Characteristics of 3 Different Sizes of Similar Fans at 3400 rpm

For example, in one investigation of the size factor, Fan Law 1 was utilized by holding rpm (revolutions per minute) constant and varying size over the range shown in Fig. 9. Similarly, the effect of rpm was studied by varying the speed of a given size fan. Other fan laws were used to investigate other variables.

Sound Laws for Fans

Based on tests as just described, the change in sound-power level of a series of homologous fans were found to be a function of fan size, speed, capacity and pressure as shown in the columns labeled Sound Laws for Fans in Table 1. These laws are intended to be applied to the overall sound-power level, but may, with limitations, be applied to octave-band analyses. These limitations are concerned primarily with the blade frequency. Octave-band analyses made under the conditions of these tests show that the peak sound power is in the octave band in which the blade frequency occurs. If the fan speed is changed to such an extent that the blade frequency is shifted to a different octave band, it is necessary to move the peak sound power to this octave in the frequency spectrum.

These laws have been found to hold for centrifugal and axial fans over operating ranges normally encountered in commercial and industrial use. Propeller fans have agreed with the aforementioned laws on an overall sound-power-level basis, but have shown some deviation on an octave-band power-level basis. This deviation has occurred between calculated and measured values in the lowest octave band of large (72 in. diam.) fans. The measured values are several db higher than

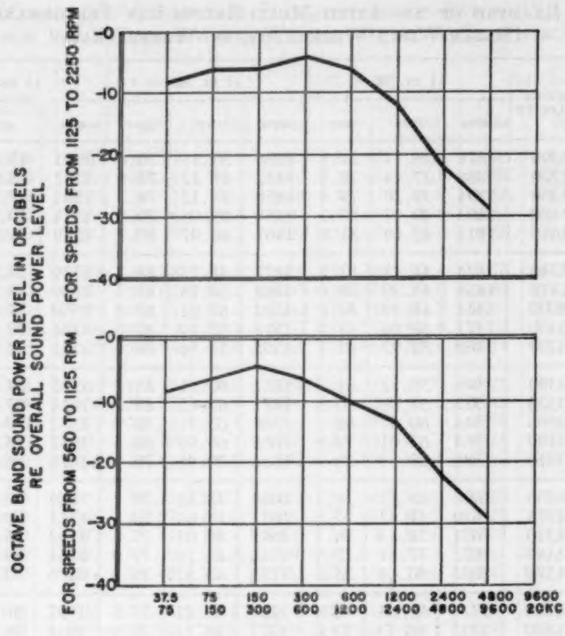


FIG. 10—CORRECTIONS TO BE MADE TO OVERALL SOUND-POWER LEVEL TO OBTAIN APPROXIMATE OCTAVE BAND SOUND-POWER-LEVELS

the calculated values. Since, due to maximum permissible stresses, these fans operate at relatively low speeds, it is believed that this low frequency component is due to large vortices from the blades.

As emphasized, the Sound Laws for Fans can be used to make calculations for equivalent points of rating only and, since the fan sound-power level is not constant for all points of rating on the performance curve, it is necessary to determine sound-power level at the point of rating at which the fan is to be used.

Sound-power-level curves have been plotted on Figs. 6, 7 and 8 to show the variation of sound-power level with different points of rating and also the variation with equivalent points of rating. The "A" and "B" points are equivalent points of rating.

The question of the variation of sound-power level as a function of horsepower has been raised in the past and conflicting statements have been made as to the proper relationship. It may be seen that when one of the independent variables is horsepower, there are 4 different values of the horsepower ratio by which sound-power level must vary. Selection of the proper ratio depends upon whether the other independent variable is capacity, pressure, size or speed. However, Fan

Table 2—Example of Suggested Multi-Rating Fan Performance Table
(Single Width Wheel, 36½ in. Wheel Diam.)

cfm	Outlet Velocity	11 in. SP			12 in. SP			13 in. SP		
		rpm	hp	eff	rpm	hp	eff	rpm	hp	eff
16000	2090	1379	35.71	77.5	1436	39.41	76.6	1491	43.17	75.7
17000	2220	1386	37.44	78.5	1442	41.22	77.8	1497	45.11	77.0
18000	2350	1394	39.20	79.4	1450	43.12	78.7	1503	47.11	78.1
19000	2480	1403	40.97	80.2	1458	45.06	79.5	1511	49.18	78.9
20000	2610	1413	42.69	81.0	1467	46.97	80.3	1519	51.28	79.7
21000	2740	1424	44.45	81.7	1477	48.75	81.1	1529	53.33	80.5
22000	2870	1438	46.39	82.0	1488	50.78	81.7	1539	55.38	81.2
23000	3000	1454	48.49	82.0	1502	52.91	82.0	1550	57.47	81.8
24000	3130	1471	50.66	81.9	1518	55.19	82.0	1564	59.80	82.0
25000	3260	1488	52.92	81.7	1535	57.56	81.9	1581	62.29	82.0
26000	3390	1506	55.32	81.3	1552	60.01	81.7	1597	64.85	81.9
27000	3520	1525	57.80	80.8	1571	62.62	81.3	1614	67.50	81.7
28000	3660	1544	60.37	80.2	1589	65.31	80.9	1633	70.31	81.4
29000	3780	1564	63.01	79.6	1608	68.09	80.3	1652	73.22	80.9
30000	3910	1586	65.72	78.9	1628	70.96	79.7	1670	76.22	80.4
31000	4050	1607	68.52	78.2	1649	73.88	79.1	1689	79.32	79.9
32000	4170	1629	71.42	77.5	1671	76.90	78.5	1711	82.45	79.3
33000	4310	1651	74.43	76.7	1692	80.03	77.8	1732	85.70	78.7
34000	4440	1672	77.41	75.9	1714	83.26	77.0	1754	89.05	78.0
35000	4570	1693	80.48	75.2	1735	86.53	76.3	1776	92.51	77.3
36000	4700	1714	83.65	74.4	1756	89.82	75.6	1797	96.07	76.6
37000	4830	1735	86.93	73.6	1777	93.22	74.9	1818	99.59	75.9
38000	4960	1756	90.31	72.8	1798	96.73	74.1	1839	103.22	75.2
39000	5100	1779	93.94	71.8	1819	100.34	73.3	1860	106.96	74.5
40000	5220	1803	97.70	70.8	1841	104.10	72.5	1881	110.81	73.8

Law 5 states that horsepower varies as capacity multiplied by pressure while sound-power level varies as log (capacity multiplied by pressure squared). Therefore, since either of these values may be chosen at random, the sound will not vary in any prescribed manner based upon horsepower using other fan laws.

Specific Power Level

As pointed out earlier, specific speed is used to describe the characteristics of a series of fans, and in a similar manner the arbitrary term *specific-power level* is used to relate noise characteristics of a geometrically similar series of fans. This term is defined as the power level a similar fan would make when operated at 10,000 cfm, and 1 in. static pressure. A typical specific-power-level curve is shown in Fig. 5, plotted against capacity. When the point of rating is known for a particular application, the power level of a fan selection made from this series of fans can be determined by

$$\mathrm{PWL}_s = K_s + 10 \log [(\mathrm{cfm} \times P^2)/10{,}000] \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

PWL_s = sound-power level of fan selected.

K_s = specific-sound-power level at equivalent point of rating.

TABLE 2 (Continued)—EXAMPLE OF SUGGESTED MULTI-RATING FAN PERFORMANCE TABLE (SINGLE WIDTH WHEEL, 36½ IN. WHEEL DIAM.)

CFM	OUTLET VELOCITY	14 IN. SP			15 IN. SP			16 IN. SP		
		RPM	HP	EFF	RPM	HP	EFF	RPM	HP	EFF
16000	2090	1544	47.01	74.9	1596	50.90	74.1	1645	54.85	73.4
17000	2220	1550	49.07	76.2	1601	53.10	75.5	1651	57.18	74.8
18000	2350	1555	51.17	77.4	1606	55.33	76.7	1656	59.54	76.0
19000	2480	1563	53.36	78.4	1612	57.60	77.8	1661	61.93	77.2
20000	2610	1571	55.60	79.2	1620	59.97	78.6	1668	64.40	78.1
21000	2740	1579	57.88	79.8	1628	62.38	79.4	1676	66.95	78.9
22000	2870	1589	60.05	80.6	1637	64.80	80.0	1684	69.53	79.6
23000	3000	1599	62.27	81.3	1647	67.14	80.8	1693	72.08	80.3
24000	3130	1609	64.53	81.8	1657	69.52	81.4	1703	74.58	80.9
25000	3260	1625	67.08	82.0	1668	71.96	81.9	1714	77.14	81.5
26000	3390	1641	69.76	82.0	1684	74.75	82.0	1725	79.79	82.0
27000	3520	1658	72.52	81.9	1700	77.62	82.0	1742	82.79	82.0
28000	3660	1675	75.38	81.7	1717	80.59	81.9	1758	85.87	82.0
29000	3780	1693	78.40	81.4	1734	83.65	81.7	1775	89.04	81.9
30000	3910	1712	81.53	81.0	1753	86.89	81.4	1792	92.32	81.7
31000	4050	1731	84.75	80.5	1771	90.24	81.0	1810	95.78	81.4
32000	4170	1750	88.07	80.0	1790	93.68	80.5	1829	99.35	81.0
33000	4310	1771	91.44	79.4	1809	97.23	80.0	1848	103.02	80.6
34000	4440	1792	94.90	78.8	1830	100.83	79.5	1867	106.79	80.1
35000	4570	1814	98.48	78.2	1851	104.53	79.0	1887	110.63	79.6
36000	4700	1836	102.18	77.5	1873	108.34	78.4	1909	114.56	79.0
37000	4830	1858	105.98	76.8	1894	112.26	77.7	1930	118.60	78.5
38000	4960	1879	109.78	76.2	1916	116.31	77.0	1952	122.77	77.8
39000	5100	1899	113.65	75.5	1938	120.40	76.4	1974	127.06	77.2
40000	5220	1920	117.62	74.8	1958	124.50	75.8	1995	131.44	76.5

cfm = cubic feet per minute of fan selection.
P = static pressure of fan selection.

The specific sound-power-level curve can be valuable in fan selection where sound is an important consideration. It will be noticed in Fig. 5 that the actual sound-power-level curve is reasonably flat in the region normally used, and there is the mistake of inferring that such a fan may be rated anywhere in that region with equally good results. The specific sound-power curve, however, shows this reasoning to be false, and the fan selection should be made at the minimum value of the specific sound-power curve if the selection is made on noise alone.[4]

PRESENTATION OF DATA

While the sound laws do not present an unduly complicated computation for the design engineer, it is felt that some simplification of data can be made so that the sound-power level of a fan selection may be readily calculated.[2d] One method of presenting these data to engineers is shown in Figs. 5 and 10 and Table 2. Table 2 is a page from a typical multi-rating table for one size of a geometrically similar

series of fans. This page is similar to the multi-rating tables now in use, except that in addition to the values of rpm and horsepower for various values of cfm and static pressure there has been added an efficiency figure for the fan at that point of rating.

A typical fan performance curve, Fig. 5, would be made a part of this table and would include specific sound-power-level values for this fan. Efficiency is constant for a given point of rating so that the efficiency value from the table may be used as a reference point to establish the specific-sound-power level of the fan under the particular catalog conditions of interest. Once the specific-sound-power level has been determined, the actual sound-power level is calculated from Equation 3.

Further simplification could be made by the fan manufacturer for applications of Sound Law 1 since the appropriate sound-power-level corrections could be worked out for the standard catalog sizes of that fan. The engineer would find it necessary to work out only the speed correction.

In addition to these overall sound-power-level values it would be necessary to include a typical octave-band analysis for this particular series of fans. Information of this type will give the engineer information on the characteristics of the sound for fan noise evaluation.

Octave-band sound-power-level data should be included as a part of the multi-rating tables and may be presented as shown on Fig. 10. Actual values cannot be given but rather correction values to be applied to the overall sound-power level to obtain the values for each band. It will probably be necessary to give values for several speed ranges as shown.

SPECIAL COMMENTS

A word of explanation should be added for those engineers who are unfamiliar with the *sound-power-level* concept. This approach to the evaluation of equipment noise results in absolute numerical values that are higher than those decibel values engineers are accustomed to reading when considering overall *sound-pressure* levels. These sound-pressure levels as reported in the past are now considered to be inadequate for an engineering evaluation of equipment noise. In transferring to the sound-power-level concept, a new framework of reference is necessary, and the engineer must become accustomed to the new and higher numerical values for the equipment he is using.

It should also be pointed out that an attempt has been made here to relate changes in the sound power of fans to changes in fan operating characteristics, and these considerations are limited to this field. There are other sources of noise in fan assemblies, such as unbalance, bearing noise, structural resonance, motor noise, coupling noise and belt noise, but these have not been considered as a part of this problem.

GENERAL COMMENTS

The problem of ventilation system noise is a broad and complex one and, since evaluation of fan noise is an important factor in overall system noise, it is hoped that the information given in this paper will be helpful to design engineers in evaluating fan noise under the particular conditions of interest to them.

REFERENCES

1. ASHAE RESEARCH REPORT No. 1609—Evaluation of Four Methods for Determining Sound-Power Output of a Fan, by W. F. Kerka (ASHAE TRANSACTIONS, Vol. 63, 1957, pp. 367–88).

2. Papers presented at Symposium on Sound and Vibration, Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 26, 1957, published in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, as follows: *a.* Noise Control Problems in Air-Conditioning Equipment, by R. J. Wells (August 1957, p. 138); *b.* Techniques of Sound-Power-Level of Unitary Equipment, by R. N. Hamme (September 1957, p. 143); *c.* Sound Standards for Testing and Rating, by H. C. Hardy (September 1957, p. 148); *d.* Application of Sound Standards to Equipment, by R. E. Parker (September 1957, p. 154); *e.* Estimating Octave Band Levels of Noise Generated by Air-Conditioning Systems (September 1957, p. 160). These same papers are available from the Society in a separately bound bulletin.

3. *Handbook of Noise Control*, Chapter 25—Fan Noise (McGraw-Hill Book Co., Inc., New York, N. Y.).

4. *Fan Engineering*, by R. D. Madison (published by Buffalo Forge Co., Buffalo, N. Y.).

5. *Acoustics*, by L. L. Beranek (McGraw-Hill Book Co., Inc., New York, N. Y.).

6. *Sound and Vibration*, by P. M. Morse (McGraw-Hill Book Co., New York, N. Y.).

APPENDIX

SOUND-POWER LEVEL VALUES

In a reverberant field the sound-power-level, PWL, is related to the sound-pressure level, SPL, by:

$$\text{PWL} = \text{SPL} + 10 \log R - 6.0$$

where

$$R = \text{room constant}$$

If a calibrated noise source is used as a comparison standard, the sound-power level of an unknown source may be determined as follows:

PWL_c = sound-power level of calibrated source in db re 10^{-13} watts.
PWL_x = sound-power level of unknown source in db re 10^{-13} watts.
SPL_c = sound-pressure level of calibrated source in db re 0.0002 microbar.
SPL_x = sound-pressure level of unknown source in db re 0.0002 microbar.
$PWL_c = SPL_c + 10 \log R - 6.0$
$PWL_x = SPL_x + 10 \log R - 6.0$

Since $10 \log R - 6$ appears in both expressions it may be eliminated to give:

$PWL_x = PWL_c - SPL_c + SPL_x$

FAN LAW AND SOUND LAW CALCULATION

(a) Fig. 6 shows the effect of reducing the speed by 100 rpm.

(b) The original point of rating on the base curve is shown as the "A" points.

Capacity	13,000 cfm.
Static pressure	1.25 H_2O
Horsepower	3.56 hp
Sound-power level	98.5 db
Speed	600 rpm

(c) To determine these quantities at 500 rpm, Fan Law 1 and Sound Law 1 are used and since, in this case no size change is involved, the size ratio is 1.

$$\text{Capacity} = 13,000 \times (1)^3 \times \left(\frac{500}{600}\right) \times 1$$

$$= 10,800 \text{ cfm.}$$

$$\text{Static pressure} = 1.25 \times (1)^2 \times \left(\frac{500}{600}\right)^2 \times 1$$

$$= 0.87'' \ H_2O$$

$$\text{Power} = 3.56 \times (1)^5 \times \left(\frac{500}{600}\right)^3 \times 1$$

$$= 2.07 \text{ hp.}$$

$$\text{Sound-power level} = 98.5 + 70 \log (1) + 50 \log \frac{500}{600} = 92.3$$

$$= 92 \text{ (accuracy justifies only to nearest db)}$$

These calculated points are shown as points B on Fig. 6.

USE OF MULTI-RATING TABLES

(a) To compute the sound-power level of a point of rating of a fan selected from multi-rating tables, use is made of the fan efficiency and specific sound by the following method.

(b) Assume a system requirement of 30,000 cfm at a static pressure of 12 inches of water. A selection from Table 2 gives:

1628 rpm
70.96 hp
79.7 percent efficiency

(c) Since efficiency and specific sound are constant for a given point of rating, the efficiency figure may be used to determine the specific sound at this rating point. A characteristic curve for the fan rated in Table 2 is shown on Fig. 5. Locating the 79.7 percent point on the efficiency curve (Point 1) gives a corresponding point on the specific sound curve of 93 db (Point 2).

(d) To find the sound-power level at the required conditions, Equation 3 is utilized, thus

$$PWL_s = K_p + 10 \log \frac{\text{cfm} \times P^2}{10{,}000}$$

$$= 93 + 10 \log \frac{30{,}000}{10{,}000} \times 12^2 = 119 \text{ db.}$$

(e) To determine approximate sound-power levels in each octave band, use is made of the upper curve of Fig. 10.

OCTAVE BAND	DB CORRECTION TO OVERALL PWL	ESTIMATED PWL BY OCTAVE BAND
37.5–75	−8	111
75–150	−6	113
150–300	−6	113
300–600	−4	115
600–1200	−6	113
1200–2400	−12	107
2400–4800	−22	97
4800–9600	−29	90

DISCUSSION

J. B. CHADDOCK*, Troy, N. Y. (WRITTEN): The authors have prepared an interesting and informative paper on the use of the *Fan Laws* in predicting changes in the sound-

* Associate Professor, Rensselaer Polytechnic Institute.

power level of a fan when its operating conditions are changed. The question I would like to raise here is whether these *Fan Laws* are really laws, or whether they are based on an empirical equation obtained from a very limited amount of data.

In Chapter 40 of the 1957 HVAC GUIDE the following equation is given for calculating the sound-power level of a centrifugal or axial flow fan

$$\text{PWL} = 100 + 10 \text{ Log } P + 10 \text{ Log } p \quad \text{(A)}$$

where P is the brake horsepower of the fan motor, p the total fan pressure in inches of water, and PWL = 10 Log $(W/10^{-13})$ if W is the acoustic power in watts. Equation (A) can also be written as

$$W = Pp/1000 \quad \text{(B)}$$

This says that the acoustic output of a fan is proportional to power times pressure or, alternatively, air quantity times pressure squared, *i.e.*

$$W = Qp^2 \times \text{constant} \quad \text{(C)}$$

Dimensional analysis applied to a fan gives the following dimensionless groups

$$\frac{Q}{ND^3} \text{ and } \frac{p}{\rho N^2 D^2}$$

where N is the fan speed, D the characteristic fan size, and ρ the air density. All *Fan Laws* relating power, size, speed, pressure, air quantity, and density are derived from these 2 dimensionless groups. Application of these dimensionless groups to Equation C results in the sound power of a fan having the following proportionality

$$W \sim \rho^2 N^5 D^7 \quad \text{(D)}$$

Equation D corresponds to the *Fan Laws* used by the authors in this paper. Hence, it is seen that these laws agree with Equation A taken from the HVAC GUIDE.

Equation A was originally presented in a paper in the *Journal* of the *Acoustic Society of America* by Beranek, Kamperman, and Allen, as PWL = 100 + 10 Log P. In a later paper by Allen in *Noise Control* the equation was modified to include the pressure term. He stated that, the inclusion of the pressure term showed improved fit with experimental data. It is significant that all the data leading to Equation A was made on fans operated near rated capacity, and is not intended to predict the sound-power level of a fan operated at other conditions. In fact, it is stated in the paper by Beranek, Kamperman, and Allen that, on the basis of data collected by Piestrup and Wesler at MIT., the sound power of a *given fan* under various operating conditions will be more nearly proportional to 20 Log P. Fig. A, shown here, is a reproduction based on Fig. 3 in the paper by Beranek, *et al.* The dotted line is the correlation of many fans operated at rated load and corresponds to Equation A, while the 2 solid lines are for 2 commercial centrifugal fans under varying load conditions. Both solid lines can be represented within ±2 db by the relation

$$\text{PWL} = 9 + 20 \text{ Log } P \quad \text{(E)}$$

which gives

$$W = \text{constant} \times P^2$$

or

$$W \sim \rho^2 N^6 D^{10} \quad \text{(F)}$$

Hence for a given fan, or geometric series of fans, the data of Piestrup and Wesler indicate that the *Fan Law* relating acoustic power to other fan variables is significantly different from that found from Equation A and given by the authors in this paper.

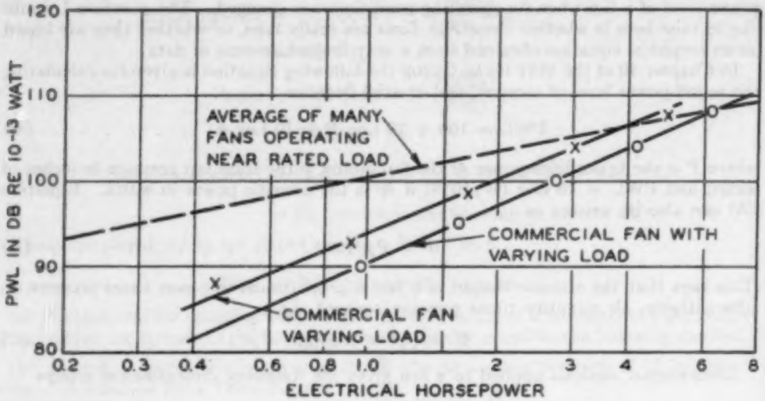


FIG. A—RESULTS OF TESTS SHOWING VARIATION OF SOUND-POWER LEVELS WITH ELECTRICAL HORSEPOWER

I wonder, then, whether there is sufficient data on fan noise at this time to say whether the acoustic power of a given fan will vary as N^5 as stated in this paper, or as N^6; also whether D^7 is more nearly correct than D^{10}?

W. F. KERKA, Cleveland, Ohio (WRITTEN): In reviewing the paper I have come to appreciate more fully the fact that an air-moving device is a complex mechanism. This paper furnishes much-needed data on the sound output of fans with respect to their operating and dimensional characteristics. This is important, since the accurate prediction of equipment noise is necessary for applying effective control, especially where it would be impossible or economically impractical to measure the acoustic output at all operating conditions.

The Fan Laws as presented in the paper are based on the fundamental laws of fluid statics and dynamics; hence the relationships between the variables and their corresponding exponents are absolute, and can be derived from a purely theoretical analysis. Therefore, it would appear that the variables and their exponents in the Sound Laws also bear this theoretical relationship. In the paper, however, the authors point out that the Sound Laws were based on tests of a number of fans. From this statement, one would conclude that the exponents were derived from experimental data. I should like to ask the authors to clarify this point.

RAYMOND MANCHA, Pittsburgh, Pa.: The authors have repeated a statement frequently made by other observers that fan sound power is minimum at or close to the fan operating point of maximum efficiency.

This statement is contrary to my many years of fan testing experience and appears to be quite illogical when viewed on the basis of airflow through a fan.

First, the authors omit definition of *Maximum Efficiency* as to whether reference is to fan static or fan total, stage static or stage total. Each of the aforementioned conditions represents a different internal flow-pattern and different acoustical power levels. Furthermore convential design practice for vaneaxial fans does not select optimum turbulence airfoil attack angles for any of the aforementioned operating points.

Viewed quantitatively, the magnitude of the fan acoustical power is very insignificant compared to the total energy dissipated within a fan so that any relationship between

minimum acoustical fan power and fan power dissipation at any of the above-mentioned efficiencies is accidental.

When considering the wide variation in volume/speed relationships for any type fan with and without Evasé stacks for maximum (static-total) efficiency operation the fallacy of the authors' statement becomes apparent.

C. M. ASHLEY, Syracuse, New York: I think that the authors are to be congratulated on making available this material on the concept of the specific sound-power levels because it has been very helpful in giving a much better idea of the relationship of the general characteristics of the fan and the sound-power level. Another small point is that I would be much happier to see the specific sound-power level based on one cfm rather than 1000. Then it would fit very nicely into the basic equation which, in general, has been recognized as 65 10 $\log_{10}$ of the air quantity and cfm times 20 $\log_{10}$ of pressure in inches.

Another point that the authors mentioned and which I would like to emphasize is that the method of testing does not measure fully the low frequency which is delivered through the fan discharge, when a duct is applied to the discharge of the fan and for this reason, it can be expected that the total sound-power level would be higher than the figures which are shown.

Also, with respect to the comment of the last discusser, I think the important factor which tends to minimize the sound power output near the point of the fan is the fact that near that point there is the minimum of separation losses and the noise definitely is associated with those separation losses.

AUTHORS' CLOSURE (Mr. Graham): The authors wish to thank the discussers for the interest shown and to express their appreciation for the helpful suggestions that have been made.

Mr. Kerka of the ASHAE Laboratory has been very active in this field and some material from his recent paper on this subject has been utilized as background for the present paper. In answer to his request for test data, included are 2 new figures in this discussion. Figs. B and C show the results of sound tests made in the authors' laboratory. These show fair agreement with the theoretical curves indicated as solid lines on the figures. Reluctance to include these in the original paper was due to the lack of a generally accepted method of fan sound-power level determination. The presentation of absolute numerical values should be done only after a code has been established.

In regard to Mr. Kerka's comment the basis of the sound laws was a rationalization of the accepted fan laws. Tests over wide ranges of size, speed, capacity and pressure are a verification of this generalization. It is an accepted fact that the intensity of sound varies as the square of the effective pressure in the sound wave. It would be rational to assume that the intensity would also vary as the square of the air pressure produced by the fan since they are motivated by the same means. For a given velocity pressure, the cfm varies as the area through which the air passes. Since sound power varies as the area times intensity it can be rationalized that sound power varies as 10 log capacity. This is the basis of law 5, from which the other laws are deduced. It is recognized that there are numerous items that cause some deviation from perfect correlation, the chief of which is probably resonances that occur at different speeds and under different conditions of set-up.

The point to remember is that the method set forth here relates the change in noise with change in performance. Absolute values of sound power will depend upon the type of fan under consideration as well as its point of rating on the performance curve. When tests on more fans are available and the technique of sound-power measurements is codified, sound-power levels should be predictable within reasonable tolerances.

Mr. Ashley has emphasized a very important point in connection with fan noise measurements and that is the difficulty of obtaining sound-power level in the low frequency ranges. This Society and other interested societies are working on this problem and it is expected that a satisfactory code will be approved in the near future. At that

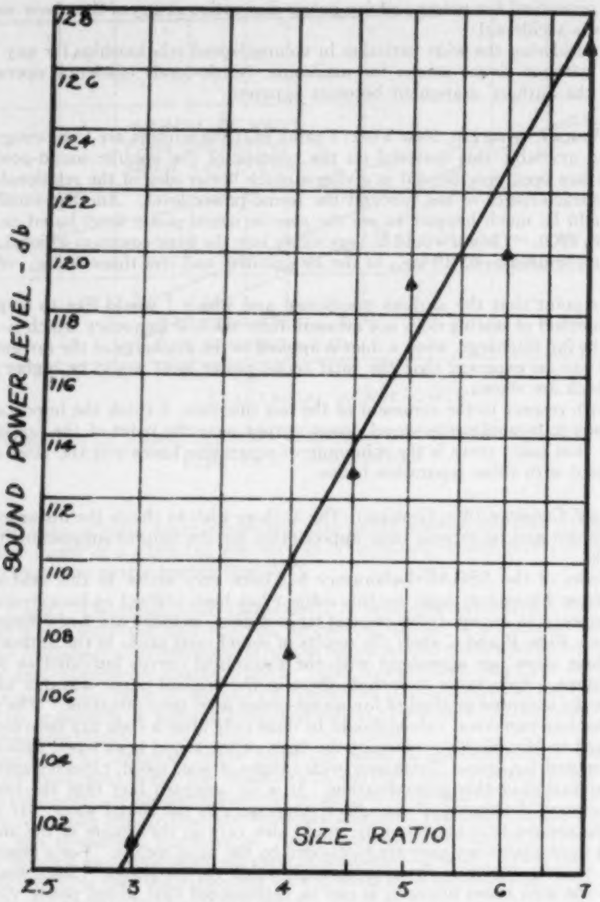


FIG. B—TEST RESULTS SHOWING RELATION BETWEEN SOUND-POWER LEVEL AND SIZE RATIO

time, absolute numerical values as here presented may change but it is believed that the relative values of sound-power levels as functions of cfm, pressure, size, speed and horsepower will remain as stated in this paper. The sound laws will be valid but as Mr. Ashley has pointed out, accurate values of PWL_b cannot be established until general agreement is reached on how to measure and report the low frequency portion of the spectrum.

The authors are glad to know that Mr. Ashley has found the specific sound power concept very helpful. The reason the value of specific sound power was referred to

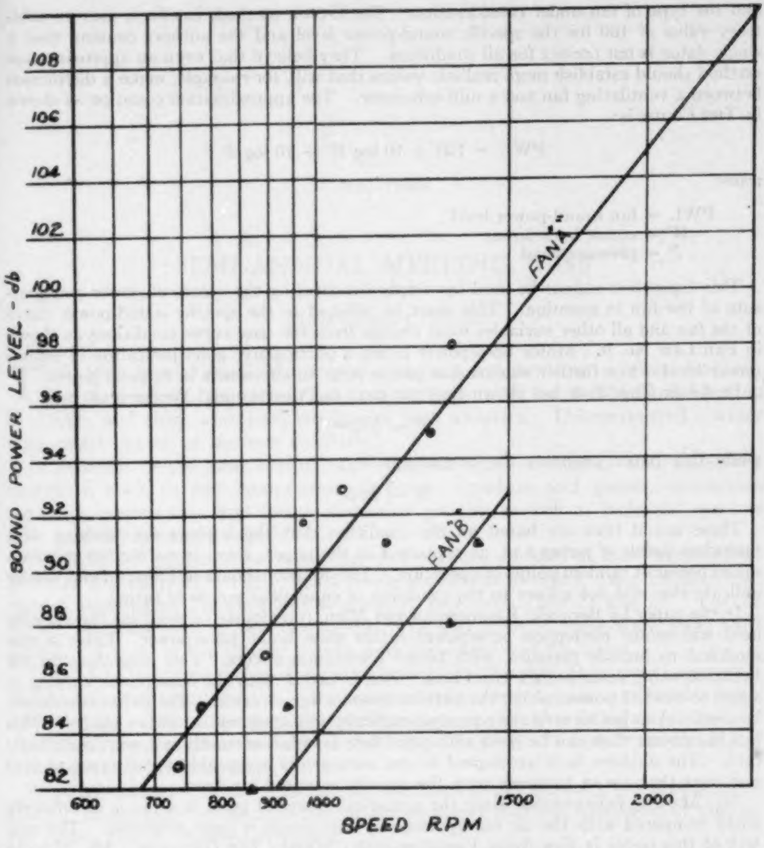


FIG. C—RELATIONSHIP BETWEEN SOUND-POWER LEVEL AND FAN SPEED FOR 2 FANS FROM AUTHORS' TESTS

10,000 cfm at 1-inch static pressure instead of 1 cfm at 1-inch pressure is that the value was a little easier to visualize. Consider a fan 5.85 in. in diameter operating at 35,700 rpm to give 1 cfm at 1-inch static pressure. This might give a sound power of, say, 50 db. It would be much easier to visualize a 36½-in. diam. fan operating at 600 rpm to give 50 plus 10 log (10,000) equals 90db sound power. This is also the reason the decibel unit is used instead of the unit, bel.

Professor Chaddock has made a comparison of the authors' method of calculating sound-power level with the approximate method given in the 1957 GUIDE. The authors believe their method has a number of advantages over the GUIDE method. In this paper the authors tried to emphasize the importance of *point of rating* and *specific sound-power level.* These factors take into account the operating conditions of the fan

and the type of fan under consideration. The GUIDE method, in effect, uses an arbitrary value of 100 for the specific sound-power level and the authors contend that a single value is not correct for all conditions. They believe that even an approximation method should establish more realistic values that will, for example, make a distinction between a ventilating fan and a mill exhauster. The approximation equation as shown in THE GUIDE is:

$$\text{PWL} = 100 + 10 \log W + 10 \log P$$

where

PWL = fan sound-power level
W = motor horsepower
P = pressure head

This expression cannot be used by simply substituting the values of power and pressure of the fan in question. This must be referred to the specific sound-power curve of the fan and all other variables must change from the base curve conditions as shown in Fan Law No. 9. Motor horsepower is not a particularly good indication of sound-power level. For further explanation please refer to comments in original paper.

Professor Chaddock has shown that the data of Priestrup and Wesler give:

$$W \sim N^5 D^{10},$$

while this paper proposes the proportionality:

$$W \sim D^5 D^7$$

These sound laws are based on the condition that the authors are working with ***equivalent points of rating*** and, as explained in the paper, there is no way to calculate sound power at random points of operation. The additional data of Piestrup and Wesler indicate they did not adhere to the condition of equivalent points of rating.

In the paper by Beranek, Kamperman and Allen, (previously referred to) the variable used was motor nameplate horsepower rather than brake horsepower. Later it was modified to include pressure, with better correlation effects. Two objections to the Beranek, *et al.*, reference are cited here. One is tied in with the horsepower varying as speed to the 5/2 power, which the authors know is not accurate. The cube ratio should be used. Another form of the equation originally included number of fan blades. This is a parameter that can be used with axial flow fans but certainly not with centrifugal fans. The authors have attempted to use parameters acceptable to all types of fans and ones that are in harmony with the usually accepted fan laws.

Mr. Mancha believes that since the acoustical power of noise in a fan is so infinitely small compared with the air horsepower any correlation is *quite accidental*. The subject of this paper is *Fan Noise Variation with Changing Fan Operation*. Mr. Mancha would certainly agree that there is a very definite change in horsepower as size and speed are varied. Why would he not suppose that fan noise had some systematic change instead of some accidental change? Mr. Mancha and Dr. Chaddock both seem to think the authors are trying to prove Equation 1 of the latter's comments. The latter part of this equation 10 log P + 10 log P is the *change* the authors are concerned with and agrees with sound law (9) of the authors' paper. The first part of the equation, namely the constant 100, is obviously not suitable for all types of fans and this is the part for Mr. Mancha to criticize and take issue with. In doing so the authors concur as it has been their experience that this *constant* can vary as much as 10 to 15 db depending upon fan types, the more efficient types generally having the lower *constant*.

The authors repeat that generally the fan noise is lowest at or near the point of maximum efficiency. More specifically, one will find that the specific sound power based on static pressure is a minimum near the point of maximum static efficiency and the specific sound power based on total pressure is a minimum near the point of maximum total efficiency.

1639

No. 1639

SEMI-ANNUAL MEETING, 1958

MINNEAPOLIS, MINN.

THE 1958 Semi-Annual Meeting was marked by numerous unusual features. One of these was that it was held concurrently with the 54th Annual Meeting of *ASRE*, and there were joint sessions of both societies. This permitted a wider than usual choice of sessions to attend.

In addition to the joint events, each program retained most of the established functions, such as golf competitions open to members and guests, committee meetings, conferences, and forum discussion groups, as well as technical sessions with formally prepared papers.

The ASHAE program was made up of 5 sessions all held at the Pick-Nicollet Hotel in Minneapolis. There was a single session on Monday, June 23, and concurrent sessions on Tuesday and Wednesday mornings, June 24 and 25. Two of these 5 sessions were of the symposium type and both of these were joint with *ASRE*. A total of 9 technical papers were presented and discussed.

The *ASRE* program included a general assembly, 3 technical sessions, numerous discussion groups and forums. Programmed also were 2 joint conferences with ASHAE, one of these being on air conditioning and the other being on education. The *ASRE* program also was held on the 3 days, June 23, 24 and 25, at the Hotel Leamington in Minneapolis.

With the incentive of the 2 concurrent meetings, many were attracted to Minneapolis. A summary of the registration figures shows that 384 members, 67 guests, 123 ladies, and 27 children attended.

Each society had the benefit of the efforts of its own Committee on Arrangements, and the success of the meetings was largely due to the work of these committees.

Meeting in the International Room at 9:00 a.m. on Monday, June 23rd, the first session was opened by Pres. E. R. Queer, University Park, Pa. J. S. Locke, general chairman of the Committee on Arrangements, outlined the program features, and extended the greetings of the Minnesota Chapter.

President Queer then appointed the following 3 members to serve as the Committee on Resolutions: S. F. Gilman, Syracuse, N. Y.; E. K. Wagner, Philadelphia, Pa.; and P. E. McNall, Jr., Hopkins, Minn.

Three papers which were on the program (see page 347) for the session were presented and discussed.

President Queer then announced that several proposed By-Law changes would be read. He called on A. V. Hutchinson, executive secretary, New York, N. Y.,

to present the By-Law information. Mr. Hutchinson read the proposed wording of changes to ARTICLE IV, *Section 6 (a)*; to ARTICLE VII, *Section 3 (e)*; to ARTICLE VIII, *Section 3*; and to ARTICLE IX, Section 1. The wording as read was that approved by the Council at its meeting on April 20, 1958. President Queer explained that it was necessary to present these proposed amendments so that they might be voted on at the next Annual Meeting of the Society. No additional amendments to the By-Laws were proposed. Following is a full statement of these proposed amendments.

ARTICLE IV. ***Section 6.*** **Investment of Funds.**

(*a*) The Society Reserve Fund and such other funds as may be allocated by the Council for investment may be invested and reinvested, without restriction, in any kind of property, real, personal or mixed.

ARTICLE VII. ***Section 3.*** **General Committees.**

(*e*) *Honors and Awards Committee*, consisting of three (3) MEMBERS who have been MEMBERS in good standing for not less than ten (10) years, have served on the Council, and who are well acquainted with the ideals of the Society and the activities, prominence, and prestige of its Members. Subject to the direction of the Council, the said committee shall be responsible for the administration of all Honors and Awards granted by the Society; the review, interpretation, and revision of policy with respect thereto; the acceptance of grants for the establishment of awards; and the creation of new awards which will assist in attaining the purposes and objectives of the Society.

ARTICLE VIII. ***Section 3.*** **Nominating Committee.**

. . . Alternate members may be present only at the organization meeting of the committee at the Annual Meeting but shall not participate in the deliberations thereof or vote therein. No alternate member may be present at subsequent committee meetings except in the absence or disability or resignation of the member or the first alternate of the one of the eight (8) groups with which he was selected. The absence, disability or resignation of a member or an alternate member eligible to participate shall serve automatically to terminate his membership on said committee. . . .

ARTICLE IX. ***Section 1.*** **Prerequisites.** These By-Laws may be amended by a two-thirds vote of the Society at an Annual Meeting or Semi-Annual Meeting thereof, provided that written notice of the proposed amendment, . . .

President Queer also announced that a statement would be made on Wednesday morning concerning the status of the proposed merger with *ASRE*, and following this, he adjourned the session.

First Vice Pres. A. J. Hess, Los Angeles, Calif., opened the second session in the Junior Ballroom at the Pick-Nicollet at 9:00 a.m. on Tuesday, June 24th. The authors of the three papers programmed for this session were introduced and presented their papers, each of which was discussed.

Following the discussion of the third paper, Vice President Hess adjourned the session.

The third session, held at 9:00 a.m. on Tuesday June 24th, was called to order by Past Pres. P. B. Gordon, New York, N. Y., in the absence of 2nd Vice Pres. Walter A. Grant, Syracuse, N. Y. It was a symposium type session on the subject of Thermal Insulation, and was a joint session with *ASRE*.

Mr. Gordon introduced P. N. Vinther, Dallas, Tex., symposium manager, and M. W. Keyes, Pittsburgh, Pa., moderator for the session. Five speakers had prepared papers especially for this session. The moderator, Mr. Keyes, called successively on their authors and then announced that the session was open for dis-

cussion. He also stated that in accordance with the usual practice of the Society, the papers at this session and their discussions will not be printed in the TRANSACTIONS. [Substantially the full text of each paper is published in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, in the following locations: the paper by William Turbeville, July 1958 issue, beginning on p. 126; the paper by N. B. Hutcheon, August 1958 issue, p. 150; the paper by R. B. Crepps, September 1958, p. 161; the paper by W. P. Ellis, July 1958, p. 136; paper by C. G. Collins and G. W. Pomeroy, August 1958, p. 129. The discussions which took place at the session are not printed in the JOURNAL.]

The moderator also announced that a symposium bulletin will be available containing the text of all 5 of the papers together with a digest of the discussions.

At the conclusion of the discussion period and the announcements, Mr. Keyes turned the session back to Past President Gordon who thereupon adjourned it.

At 9:00 a.m. Wednesday June 25th in the International Room at the Pick-Nicollet Hotel, the fourth session was opened by Treas. J. H. Fox, Toronto, Ont., Canada. This was a joint session with *ASRE*, was of the symposium type and on the subject of Condensing Methods.

Treasurer Fox introduced W. G. Hole, Montreal, Que., Canada, who was the symposium manager and then introduced 2nd Vice Pres. A. J. Hess who served as the moderator.

Mr. Hess announced that although the printed program showed that a series of 4 papers would be presented, it had been decided that a paper prepared by R. C. Edwards, Pompton Plains, N. J., should also be presented. The title of Mr. Edward's paper is: *Air Condensers with Gravity Flow.*

The moderator asked each of the 5 speakers to present his material and following this, the session was open for discussion.

The moderator announced that it is not expected that the papers and the discussions will be printed in the TRANSACTIONS, but portions of them may appear in the JOURNAL SECTION. [Substantially the full text of the 5 papers, but without discussion, is printed in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*. The paper by John Engalitcheff begins on page 145 of the August 1958 issue; the paper by D. D. Wile on page 153, August 1958; O. J. Nussbaum, p. 155, September 1958; and the one by J. L. Wolf, p. 135, August 1958.]

The moderator also announced that a symposium bulletin will be available containing the text of all the papers together with a digest of the discussions.

Following this, Treasurer Fox turned the session back to President Queer.

At this time, President Queer made a statement concerning the status of the proposed merger with *ASRE*.

PRESIDENT QUEER'S STATEMENT CONCERNING MERGER

He started by stating that before any merger could take place all members will have an opportunity to ballot on the proposal. He then reviewed the background of the proposed merger, mentioning that a suggestion for merging was made early in the 1930's. The subject arose again and while no merger took place a cooperative plan for the exchange of publications of both Societies became effective in 1941 and still continues in effect. In 1954 a Joint Committee was established to explore ways for closer cooperation.

When the ASHAE regional plan for chapter operations was put into effect, the merger proposal was entered as a chapter agenda item from Regions 6 and 7.

Subsequently, the Regions Central Committee processed this item. In February 1957 the ASHAE Council passed a resolution instructing the Society representatives on the Joint Committee on Cooperation to broaden the scope of its work to (1) include a study of relevant facts relating to the respective percentage of membership which might be interested in a merger plan, and (2) to explore the possible plans for future consideration. Finally, at its January 26, 1958 meeting, the ASHAE Council voted that the ASHAE-*ASRE* Plan, dated January 26, 1958, be presented to the membership of the Society through the Chapters Regional Committee meetings held during the spring of 1958, and that the Society members on the Committee be authorized to proceed with further detailed study on the proposed merger.

To put this authorization by the Council into effect, President Queer had sent a letter outlining the proposed merger plan to all chapters and regional directors under date of February 17, 1958. In that letter he had expressed the desire of the Council that full and free discussion of the plan take place among the membership in the Chapters. The letter also outlined the procedure for doing this and at the same time outlined the tentative time schedule which appeared to be desirable and necessary in order to get prompt action on the plan and to meet JOURNAL contractural obligations.

As to the necessity for prompt action, President Queer pointed out that the subject has been under more or less continuous discussion since 1954 and under active discussion since January 1957. He felt, therefore, that action should be taken as promptly as possible because, as mentioned in his February 17 memorandum, many important decisions are being held in abeyance awaiting the merger action. Among these are (a) a plan to erect a new and expanded facility for Society research, (2) a new organization plan proposed by the Long-Range Planning Committee, and (3) the non-research activities of the Society had a deficit last year and are operating on deficit financing this year. Though operating surpluses remaining from earlier years are being used for the time being, certain changes in Society financing and dues will be required.

Additionally, President Queer felt that prompt action should be taken because various other decisions, besides those mentioned, are also being held in abeyance. He also felt that since it is expected that the final details of the plan will be completed shortly, the plan itself could be discussed during September, October and November in Chapter meetings and by the members at large. It is also planned to mail to each member an outline of the proposal late in August or earlier if possible. If this could be done it would permit the calling of a Special Meeting, at which meeting the proposal can be discussed. Such a meeting could be called on or about the first of December. He noted that under the By-Laws the Council can submit the plan to the members not less than ten (10) days before the date proposed for the Special Meeting, and not more than forty (40) days. He felt that this would give each member a chance not only to vote on the question promptly, but also would permit sufficient time for adequate discussion and for any member to receive any needed information before voting. He indicated that with a two-thirds favorable vote by the members voting, either in person or by proxy, in both Societies, and approval of the New York Court, the merger could be made effective as early as the date of adjournment of the 1959 Annual Meeting. President Queer closed his statement by asking if there were any questions and indicated that the session was now open to discussion from the floor. Comments concerning the merger were

made by C. D. Potter, Providence, R. I.; S. H. Nitzberg, Wood-Ridge, N. J.; B. F. McLouth, Minneapolis, Minn.; and W. A. Schworm, Ames, Ia.

NOMINATIONS

In the absence of A. W. Edwards, Cincinnati, Ohio, chairman of the Nominating Committee, President Queer read the report of this committee, indicating that A. J. Hess, Los Angeles, Calif., was nominated for president; Walter A. Grant, Syracuse N. Y., for 1st vice president; John Everetts, Jr., Philadelphia, Pa., for 2nd vice president; and J. H. Fox, Toronto, Ont., Canada, for treasurer. Nominated for members of Council were R. S. Stover, Marshalltown, Ia.; H. A. Lockhart, Morton Grove, Ill.; W. J. Collins, Jr., Oklahoma City, Okla.; and G. W. F. Myers, St. Louis, Mo.

Nominated also for the Committee on Research were R. C. Jordan, Minneapolis, Minn.; Albert Giannini, New York, N. Y.; Maurice Nelles, Chicago, Ill.; P. H. Yeomans, Philadelphia, Pa.; and H. B. Nottage, Encino, Calif.

President Queer also called for the Report of the Committee on Resolutions, and in the absence of the chairman, S. F. Gilman, Syracuse, N. Y., the report was read by P. E. McNall, Jr., Hopkins, Minn.

RESOLUTIONS

WHEREAS, the 540 members and guests of this 1958 Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS are about to depart, and

WHEREAS, the meeting was held in Minneapolis, true star of the *nord countree* whose weather during our short stay proved there really is no water shortage in the twin city—Minnehaha area, and

WHEREAS, interesting and valuable technical papers and symposia were presented and discussed, and

WHEREAS, the Minnesota Chapter, its members, and the Committee on Arrangements have provided excellent meeting facilities, together with a wide selection of social events pleasing all, from country folk to city slicker, and

WHEREAS, the *American Society of Refrigerating Engineers*, which is holding its 54th Annual Meeting simultaneously with ours, has been so very hospitable to our members, and even agreed to print their portion of the joint program upside down,

BE IT RESOLVED, that we who have been privileged to attend this meeting express our sincere gratitude:

TO the Minnesota Chapter, its officers and members for the great effort they have expended in making this meeting a success,

TO J. S. Locke, General Chairman, the honorary and vice-chairmen, and all others on the Committee on Arrangements,

TO the ladies of the Chapter for their gracious hospitality in entertaining and assisting visiting ladies and children,

TO the authors and discussers of the papers and symposia for their technical contributions,

TO the Honorable P. Kenneth Peterson, Mayor of Minneapolis, for his Welcome Luncheon address on the attributes of his city and Minnesota in its Centennial,

TO Dr. R. C. Jordan, who so ably presided at the ASHAE-*ASRE* Joint Dinner,

TO whomever it was who was responsible for the best, and shortest, Banquet Speech in our history,

TO President E. R. Queer, the other officers, Council and Committees for their continuing efforts in advancing the aims and objectives of our Society,

TO ASRE for joining with us to provide a fine technical and social program,

TO our German scientific friends from across the sea, which group includes officers of the Heating and Ventilating Section of the German Engineers Society,

TO the young people who presented the outstanding Aquatennial Preview,

TO the industries who kindly invited us to visit their facilities,

TO the *Air-Conditioning and Refrigeration Institute* who provided fluid refreshments only a small portion of which came from Lake Minnetonka,

TO *Elvis* Roach for his dignified approach to modern music,

TO the hotels for installing air conditioning since our 1949 meeting here when the temperature both inside and out was 102 F, and which air conditioning may possibly be needed this year before summer is out—Brrh!

Respectfully submitted,

The Resolutions Committee

STANLEY F. GILMAN, *Chairman*
P. E. MCNALL
E. K. WAGNER

Following the adoption of the report of the committee, President Queer adjourned the session at 12:20 p.m.

Meeting at 9:00 a.m. in the Lakeland Room at the Pick-Nicollet Hotel on Wednesday June 25, President Queer called the fifth session to order and introduced successively the authors of the three papers on the program for the session. Each paper was presented and discussed.

President Queer announced that the closing items on the program would be taken up in the fourth session in the International Room and suggested that all present should attend that session. He then adjourned the fifth session at 11:30.

JOINT SESSIONS WITH *ASRE*

Included on the program of the ASRE were 2 joint conferences. One of these was held at 2.30 p.m., Tuesday afternoon at the Hotel Leamington under the chairmanship of W. L. McGrath, Syracuse, N. Y. The conference was devoted to the subject of *Heat Operated Refrigeration and Air Conditioning.* There were 4 speakers. The first of these was G. B. Priester, Baltimore, Md., whose subject was *The Future of Gas for Air Conditioning;* the second speaker was R. B. Smith, New York, N. Y., with the subject of *New Developments in Gas-Fired Air Conditioners for Residences;* J. F. Moore, Dallas, Tex., had as his subject *Absorption Refrigeration for Large Air Conditioning Installations;* the fourth subject was *Steam Driven Turbine Refrigeration for Air Conditioning,* and was by B. A. Dmitrieff, New York, N.Y.

The second conference, which was devoted to *Education for Careers in Refrigeration and Air Conditioning,* was held at the Leamington Hotel on Wednesday afternoon with H. M. Hendrickson, Seattle, Wash., as chairman and Past Pres. John

W. James, Chicago, Ill., as co-chairman. The subject of *The Role and Responsibility of the Educator* was covered by J. B. Chaddock, Troy, N. Y., and W. F. Stoecker, Urbana, Ill.; C. M. Ashley, Syracuse, N. Y., had the topic *What Industry Expects of the Engineering Graduate;* B. H. Jennings, Cleveland, Ohio, had been assigned the subject of *How Can We Interest High School and Junior College Students in our Industry?*; Past Pres. P. B. Gordon, New York, N. Y., spoke on *The Role and Responsibilities of Trade Associations and Technical Societies.*

These conferences on the *ASRE* program proved to be attractive features to many of the ASHAE group who were in attendance.

JOINT STATEMENT ON PROPOSED MERGER

Following is the full text of a joint statement of the Councils of ASHAE *and of ASRE issued under date of July 1, 1958.*

NEW YORK, NEW YORK—In a joint statement issued by the Councils of the American Society of Heating and Air-Conditioning Engineers and The American Society of Refrigerating Engineers it was announced they have approved in principle a method of merging the two Societies. The ASRE members in attendance at their 54th Annual Meeting authorized submission of this proposal for balloting by the ASRE membership.

E. R. Queer, ASHAE President, and Cecil Boling, ASRE President, further announced that present plans contemplate that the proposal for a merger and proxy ballots will be officially mailed to the members of both Societies in late October.

Full particulars of the merger plan will be mailed to the members of both ASHAE and ASRE by September. Ballots will be taken in person or by proxy at the 45th Semi-Annual Meeting of ASRE in New Orleans, La. on December 1, 1958 and at a Special Meeting of the ASHAE membership on December 1, 1958.

PROGRAM—SEMI-ANNUAL MEETING

Pick-Nicollet Hotel, Minneapolis, Minn.

June 23–25, 1958

Saturday—June 21

9:00 a.m. Finance Committee (*North Star Room*) J. N. Livermore, *Chairman*

10:30 a.m. Executive Committee (*Hiawatha Room*) A. J. Hess, *Chairman*

1:30 p.m. Regions Central Committee (*Rod & Reel Room*) Walter A. Grant, *Chairman*

2:00 p.m. Research Executive Committee (*Arrowhead Room*) E. F. Snyder, Jr., *Chairman*

4:30 p.m. Program and Papers Committee (*North Star Room*) John Everetts, Jr., *Chairman*

Sunday—June 22

9:00 a.m. Council Meeting (*Rod & Reel Room*)

2:00 p.m. Committee on Research (*Arrowhead Room*) E. F. Snyder, Jr., *Chairman*

3:30 p.m. Reception—Hospitality Time (*Junior Ballroom*) Light refreshments served. An opportunity to meet friends and fellow members. *ASRE* members and guests invited.

7:30 p.m. Conference of Chapter Editors and Publicity Chairmen (*North Star Room*)
E. K. Wagner, *Chairman*

8:45 p.m. Reception—Aquatennial Preview (*Leamington Hotel*)
ASRE reception, ASHAE members and guests cordially invited.

Monday—June 23

8:30 a.m. REGISTRATION (*East Room*)

9:00 a.m. FIRST SESSION (*International Room*)

Call to Order—Pres. E. R. Queer

A Water-Cooled Luminaire in a Panel-Air System, by W. F. Spiegel, Philadelphia, Pa., presented by Mr. Spiegel

Pulsations in Residential Gas Furnaces with Multiple-Port Burners, by A. A. Putnam, Columbus, Ohio, presented by Mr. Putnam

Evaluation of Air Cleaners for Air Conditioning and Ventilation, Part I—Apparatus, by K. T. Whitby, A. B. Algren, R. C. Jordan, and J. C. Annis, Minneapolis, Minn., presented by Mr. Whitby

9:30 a.m. Children's Tour

9:30 a.m. Ladies' "Get-Acquainted" Brunch (*Junior Ballroom*)

11:00 a.m. Ladies' Trip To Southdale Shopping Center

12:00 noon Welcome Luncheon (*Lakeland Room*)
Toastmaster: John E. Haines, ASHAE Past President
Guest Speaker: Hon. P. Kenneth Peterson, Mayor of Minneapolis

12:00 noon ASHAE Golf Tournament (*Golden Valley Country Club*)

1:30 p.m. Inspection Trip For the Men

1:30 p.m. Standards Committee (*Rod & Reel Room*) P. N. Vinther, *Chairman*

2:00 p.m. TAC on Combustion (*North Star Room*) M. W. McRae, *Chairman*

2:00 p.m. TAC on Hot Water and Steam Heating (*Hiawatha Room*) A. O. Roche, *Chairman*

2:00 p.m. TAC on Insulation (*Birchwood Room*) G. R. Munger, *Chairman*

2:00 p.m. TAC on Odors (*Arrowhead Room*) H. L. Barnebey, *Chairman*

2:00 p.m. TAC on Physiological Research and Human Comfort (*Chippewa Room*), John Everetts, Jr., *Chairman*

5:00 p.m. ASHAE "County Fair" (*Lafayette Club, Lake Minnetonka*)

Tuesday—June 24

8:30 a.m. REGISTRATION (*East Room*)

9:00 a.m. SECOND SESSION (*Lakeland Room*)

Call to Order—1st Vice Pres. A. J. Hess

Winter Infiltration Through Swinging-Door Entrances in Multi-Story Buildings, by T. C. Min, Auburn, Ala., presented by Mr. Min

Corrosion Inhibition on Tubes in Low-Pressure Steel Boilers, by W. A. Keilbaugh and F. J. Pocock, Alliance, Ohio, presented by Mr. Keilbaugh

Heat Gain Through Windows Shaded by Canvas Awnings, by Necati Ozisik and L. F. Schutrum, Cleveland, Ohio, presented by Mr. Ozisik

9:00 a.m. THIRD SESSION (*International Room*)

Call to Order—2nd Vice Pres. Walter A. Grant

SYMPOSIUM ON THERMAL INSULATION—Joint with *ASRE*

P. N. Vinther, *Symposium Manager*

M. W. Keyes, *Moderator*

How Research Can Improve Performance of Fibrous and Reflective Insulation, by William Turbeville

Block and Pipe Insulation, by R. B. Crepps

Vapor Problems in Thermal Insulation, by N. B. Hutcheon

Thermal Insulation for Nuclear Systems, by C. G. Collins and G. W. Pomeroy

Surfacings for Glass Fiber and Foam Thermal Insulation, by W. P. Ellis

10:00 a.m. Young Folks Bus Trip And Swimming Party

10:00 a.m. ASHAE-*ASRE* Ladies' Joint Visit to General Mills and Betty Crocker Kitchens

1:30 p.m. Inspection Trip For The Men

1:30 p.m. Nominating Committee (*Aquatennial Room, 3rd Floor*) A. W. Edwards, *Chairman*

2:00 p.m. TAC on Air Cleaning (*North Star Room*) W. C. L. Hemeon, *Chairman*

2:00 p.m. TAC on Air Distribution (*Birchwood Room*) P. A. Argentieri, *Chairman*

2:00 p.m. TAC on Sorption (*Hiawatha Room*) G. C. F. Asker, *Chairman*

2:00 p.m. TAC on Thermal Circuits (*Arrowhead Room*) S. F. Gilman, *Chairman*

2:30 p.m. *ASRE* Conference on Air Conditioning—(*Leamington Hotel*). Joint with ASHAE.

6:00 p.m. Cocktail Party—*ARI* host (*Leamington Hotel, Town Hall, Lower Lobby*)

7:15 p.m. Joint Dinner (*Leamington Hotel, Hall of States, Main Lobby*)

Toastmaster: Dr. R. C. Jordan, Head, Dept. of Mechanical Engrg., University of Minnesota

Presentations

Entertainment

Dancing

Wednesday—June 25

8:30 a.m. REGISTRATION (*East Room*)

9:00 a.m. FOURTH SESSION (*International Room*)

Call to Order—Treas. J. H. Fox

SYMPOSIUM ON CONDENSING METHODS—Joint with *ASRE*

W. G. Hole, *Symposium Manager*

A. J. Hess, *Moderator*

Cooling Tower Design and Performance, by John Engalitcheff

Evaporative Condensers, by D. D. Wile

Air Cooled Condensers, by Otto Nussbaum

Economic Evaluation of Condensing Methods, by J. L. Wolf

Report of Resolutions Committee

Unfinished Business

New Business

Adjournment

9:00 a.m. FIFTH SESSION (*Lakeland Room*)

Call to Order—Pres. E. R. Queer

Activated Charcoal for Air Purification, by H. L. Barnebey, Columbus, Ohio, presented by Mr. Barnebey

Cooling Load from Pretabulated Impedances, by Harry Buchberg, Los Angeles, Calif., presented by Mr. Buchberg

Solar Energy Utilization for Heating, Cooling, Distillation and Drying—prepared by members TAC on Solar Energy Utilization, presented by R. C. Jordan

1:30 p.m. *ASRE* Conference on Education (*Leamington Hotel*). Joint with ASHAE

1:30 p.m. Ladies' Card Party (*Candlelight Room*)

1:30 p.m. Children's Theater Party

COMMITTEE ON ARRANGEMENTS

J. S. Locke—*General Chairman* John E. Haines, F. B. Rowley—*Honorary Chairmen*

G. M. Kendrick, J. F. Siegel—*Vice Chairmen*

BANQUET: L. C. Gross, *Chairman*

CHILDREN: R. M. Jack, *Chairman*

Mmes. P. L. Anderson, C. B. Ashenfelter, J. V. Borry, C. T. Hastings, R. M. Jack, A. E. Malloy, B. L. McGeorge, T. A. Ryan, T. R. Turnham.

ENTERTAINMENT: J. L. Hanson, *Chairman*

C. L. Bensen, Mr. & Mrs. J. R. Bergan, Mr. & Mrs. J. V. Borry, Mr. & Mrs. R. L. Campbell, Mr. & Mrs. C. L. Cich, Mr. & Mrs. J. F. Cummiskey, Mr. & Mrs. R. W. Crysler, Mr. & Mrs. R. E. Cook, Mr. & Mrs. M. L. Fergestad, Mr. & Mrs. W. R. Fay, Mr. & Mrs. S. L. Furber, Mr. & Mrs. J. H. Feilzer, R. G. Gridley, Mrs. J. E. Haines, Mr. & Mrs. J. A. Jester, L. J. Krause, Mr. & Mrs. R. M. Locke, Mr. & Mrs. D. L. McGeorge, Mr. & Mrs. R. H. McGinty, J. W. McNamara, Mr. & Mrs. J. D. Menth, H. C. Mills, Mr. & Mrs. J. R. Namieson, Mr. & Mrs. W. J. Ortman, Mr. & Mrs. D. D. Patrie, Mr. & Mrs. Harold Quint, Mr. & Mrs. H. P. Roberts, Mr. & Mrs. R. J. Ruth, Mr. & Mrs. K. W. Schick, Mr. & Mrs. J. L. Skarnes, Mr. & Mrs. R. E. Sorenson, Mr. & Mrs. H. T. Sparrow, Mr. & Mrs. D. W. Stover, Mr. & Mrs. D. L. Vandergon, Mr. & Mrs. F. G. Vogt, Mr. & Mrs. C. E. Wiser, Mr. & Mrs. J. W. Wheeler.

FINANCE: V. E. Pearson, *Chairman*

Ralph Etnier, O. L. Lilja, C. T. Olson, R. A. Nelson, R. L. Peterson.

LADIES: S. L. Furber, *Chairman*

Mrs. S. L. Furber, *Co-Chairman*, Mmes. J. A. Craig, J. F. Cummiskey, G. M. Kendrick, J. S. Locke, B. F. McLouth, V. E. Pearson, H. G. Sierk, William Sturm.

PUBLICITY: W. J. Ortman, *Chairman* and J. Robert Snyder

RECEPTION: M. S. Wunderlich, *Chairman*

A. B. Algren, Russell E. Backstrom, G. B. Brown, L. F. Flagg, D. M. Forfar, J. K. Gerrish, L. L. Lierboe, B. J. Mulcahy, E. P. Palmer, E. J. O'Donnell, E. F. Snyder, Jr.

SESSIONS: William Sturm, *Chairman*

J. R. Andrews, W. W. Arndt, C. B. Ashenfelter, E. J. Baker, D. C. Campbell, L. D. Freedland, J. T. Hultgren, A. P. Keller, Ray Peterson, Julian Sjordal, R. E. Torell, A. W. Wessel.

SPORTS: B. F. McLouth, *Chairman*

Lyman Hyde, E. A. Monies, E. J. O'Donnell, C. T. Hastings, M. E. Quigley, M. B Kispert, T. A. Perry, E. R. Huckins, M. E. McLouth.

TRANSPORTATION: J. A. Craig, *Chairman*

R. H. Griffiths, H. J. Mahanns, C. E. Wiser.

1640

No. 1640

A WATER-COOLED LUMINAIRE IN A PANEL-AIR SYSTEM

By W. F. SPIEGEL*, PHILADELPHIA, PA.

THE YEAR 1951 saw the practical realization of a new concept in panel cooling when a panel-air system was installed in the 200,000 sq ft Manufacturers Life Insurance Co. building in Toronto. This installation represented an economically favorable solution, and incorporates both lights and panels in 24 percent of the interior ceiling area.

The fundamental design of the system and the luminaires was the result of an extended series of investigations by Charles S. Leopold starting over a decade ago.

In spite of several pioneer installations in Europe about 1935, the practicability of large panel-cooling systems had been seriously questioned because of the danger of surface condensation. When using water temperatures sufficiently high to be *safe*, conventional methods of computing radiation and convection transfer resulted in such large required panel areas that the use of panels for cooling appeared to be uneconomical in most areas.

Starting in 1939, several papers were published[1, 2, 3] pointing out that building structures absorb a large fraction of the radiation from normal room lighting. Subsequently, Leopold described a series of experiments[4, 5, 6] showing that a cooled panel forming part of a room surface will absorb a certain quantity of short-wave radiation from artificial light or sunlight virtually independent of the panel surface temperature. In effect, a panel will absorb more heat than conventional grey body calculations indicate.

With a more elaborate understanding of panel performance, it became practical to design *safe* systems to absorb a part of the heat load in commercial buildings. In a large building, if the losses through the perimeter are counteracted, the interior requires cooling regardless of outside temperature. It has long appeared desirable to save building space by removing part of this heat through water tubing

---

* Staff Engineer, Charles S. Leopold Engineers. Member of ASHAE.
[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

rather than air ducts. Several installations have been made using pipes embedded in the ceiling, and more recently, by attaching metal ceiling pans to coolant pipes.

Even when using the entire ceiling area, however, the problem of moisture and ventilation supply is usually met by introducing small amounts of dry air to the conditioned space. The drying is most frequently accomplished by refrigeration, and some cooling effect results as a by-product. The smallest amount of air which can be introduced to the space is determined either by the lowest economical dew-point of the air leaving the apparatus, or by the minimum requirement for odor and ventilation control. The actual heat pick-up by the air can be calculated knowing the temperature resulting from run-around or economizer coils. The remainder of the room cooling load is then assigned to the panels. Such combinations have been designated *Panel-Air Systems*.

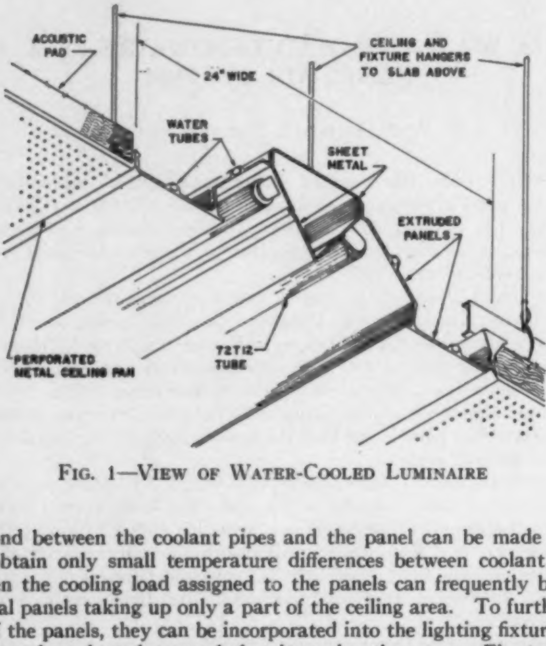


FIG. 1—VIEW OF WATER-COOLED LUMINAIRE

If the bond between the coolant pipes and the panel can be made sufficiently secure to obtain only small temperature differences between coolant and panel surface, then the cooling load assigned to the panels can frequently be absorbed by individual panels taking up only a part of the ceiling area. To further increase the effect of the panels, they can be incorporated into the lighting fixtures to intercept radiant and conducted energy before it reaches the space. Fig. 1 shows a cut view of such a luminaire. The design of these luminaires has been described in detail[6] as has the installation in the building of the Manufacturers Life Insurance Co.[7,8].

The operation of the panel-air systems was found to be unique in several ways. Low air quantities eliminated drafts. The units lent themselves to a system of control where the regulators anticipate a condition the panels will have to meet, resulting in very small deviations of room temperature throughout the day. It was also observed that the temperature tolerance of the occupants is at least as good

as with conventional methods of air distribution. It appeared that the panel-air system could take its place among conventional systems, and it became desirable to obtain accurate data on the effectiveness of the *cold light* in performing the function of cooling an occupied space.

OBJECTIVE

To be generally useful, it was attempted to describe performance in the following ways:

1. By making accurate measurements of the amount of electrical input and the heat absorbed by water flowing through the luminaire.
2. By preparing an approximate energy accounting for a typical interior area, including components of the luminaire.
3. By presenting a tabulation of the temperatures in various locations of a test area.

The use of a reduced scale model would have been costly and impractical in this case. It was felt that good data from a completed installation would furnish the most valuable information. The results stated are from readings made with instruments approaching laboratory precision.

THEORY OF INVESTIGATION

It can be assumed that the amount of heat transferred to the water flowing through the luminaire has 2 major components: one constant quantity, independent of panel surface or water temperature, and one variable quantity.

The factor most influencing the variable component would logically be a function of the difference between the water temperature at any given location on the panel and the temperature of the surrounding air and surfaces, all integrated over the entire fixture area.

Any attempt to designate a *roomside* temperature which would be an index of radiation and convection, both in the upward and downward direction, is arbitrary unless the temperature of the ceiling space is known for various conditions. The finding of a *suitable roomside index* is considered as part of the problem, rather than an assumption. Whatever it should be, the effective temperature difference can be changed by varying the supply water temperature.

The data for heat absorption by radiation from room surfaces and by natural convection were obtained by testing the fixture at night without light or air motion. This procedure eliminated effects from direct conduction, short-wave radiation, or forced convection. By similarly operating the fixture at night, but energizing the ballast and connecting the output leads to lamps in another area, the amount of heat conducted from the ballast to the water was reflected in the additional temperature rise of the water flowing through the fixture.

With the fluorescent tubes in operation and with air supplied from the diffusers, the following items contribute to the remainder of the heat absorbed by the water circulating through the luminaire:

1. Radiation by virtue of the difference between fluorescent tube surface temperature and panel surface temperature.
2. All other radiation from the fluorescent tubes striking the fixture.
3. Conduction from the tube sockets.
4. Radiation from other heated parts of the lighting fixture, such as the center pieces.

5. Increased convection resulting from the slightly warmer air in the open part of the fixture.

6. Forced convection component of heat transfer at the bottom side of the panel sections in the ceiling plane.

The radiation of the primarily short wave lengths from the fluorescent tubes should be constant. An inspection of the spectral distribution of this type of lamp shows that almost all the energy is in the visible and extremely short wave length region. A reproduction of the data given by the lamp manufacturer is shown as Fig. A-1 in the Appendix. The radiation effect from such a tube would be generally

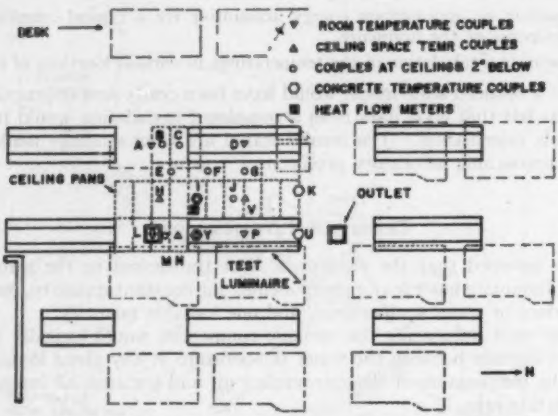


FIG. 2—PART PLAN OF 9TH FLOOR SHOWING TEST AREA

similar to that emitted by a group of grey bodies at very high temperatures, and the radiation exchange would not be materially changed by a 10 deg change of a receiver near room temperature.

The variation in the other 5 items with a change in water temperature would be expected to be small. The sum of the 6 items is thus defined as *Independent Transfer*. At the point of zero effective temperature difference, the independent transfer and direct conduction are at their minimum values and serve as an index of the effectiveness of the fixture design in intercepting the energy of the lighting at the source.

FIELD TEST

The actual test was conducted in a general office area on the 9th floor of the Manufacturers Life Insurance Company Building in a section toward the interior of the structure.

A string of 5 luminaire fixtures was chosen for the test, with the last one selected for detailed examination. A large number of measurements were made around the area of this fixture to provide a statistical survey of local heat exchanges.

Thermocouples were installed in the room and ceiling space at various heights, at locations 2 in. below the ceiling, at the metal ceiling pans, on the luminaire itself, in water wells, at the supply ducts, and at various planes in the concrete floor and ceiling slabs. Heat flow meters were installed at the floor and ceiling slab surfaces, and a radiometer fixture was devised to measure the radiation emitted from the luminaire troffer. The general arrangement of the luminaire fixtures, air outlets, and test locations is shown in Fig. 2. Detailed descriptions of the instrumentation and test procedures are given in the Appendix. A diagram of test locations at the luminaire is shown in Fig. 3.

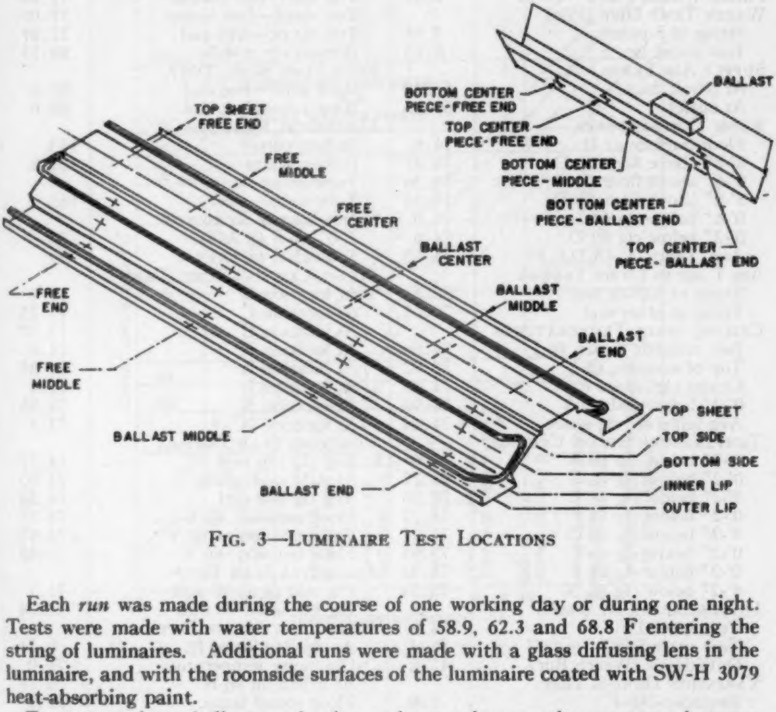


FIG. 3—LUMINAIRE TEST LOCATIONS

Each *run* was made during the course of one working day or during one night. Tests were made with water temperatures of 58.9, 62.3 and 68.8 F entering the string of luminaires. Additional runs were made with a glass diffusing lens in the luminaire, and with the roomside surfaces of the luminaire coated with SW-H 3079 heat-absorbing paint.

Tests to evaluate ballast conduction and natural convection were run between 11:00 p.m. and 2:00 a.m. following the departure of the cleaning staff. These *dark* tests were run with 55.3, 63.9 and 69.3 F entering water temperatures.

A full tabulation of 173 items for 9 runs is given in the Appendix as Table A-1. An excerpt of approximately half the readings for one run is shown in Table 1.

EVALUATION OF CONVECTION AND RADIATION TRANSFER

The heat transfer rate at each individual area was estimated by the local temperature difference, and the total transfer computed by algebraic addition.

Carroll concluded in 1948[9] that a variety of convection formulae are required to meet various applications in panel heating and cooling. Somewhat later the ASHAE Research Laboratory published a paper by Min and others[10] showing

Table 1—Excerpt of Logs Covering Test G

Panel Water Temps—F		**Light Fixture Temps**	
Ent String 5 panels	62.26	Outer lip—free middle	69.23
Ent No. 5 test panel	65.43	Inner lip—free middle	69.47
Lvg String 5 panels	66.14	Bottom side—free mid.	70.14
Panel Water Flow—GPM	1.01	Top side—free middle	71.43
Water Temp Diff (Pile)		Top sheet—free center	77.07
String of 5 panels	3.95	Top ctr pc—free end	77.97
Test panel No. 5	0.73	Bottom ctr middle	80.73
Supply Air Temps		**Fluo Tube Surf. Temp**	
At trunk duct	67.98	West tube—free end	87.0
At outlet	69.23	West tube—bal. end	88.0
Room Temperatures		**Electrical Readings**	
Floor Surface at D	74.6	Ballast volts	123
0′-2″ above floor @ D	74.27	Ballast watts	107.8
0′-6″ above floor @ D	74.36	Tube amps—west	——
5′-0″ above floor @ D	74.36	Tube volts—west	185
0′-6″ below clg @ D	74.6	**Light Meter Readings**	
0′-2″ below clg @ D	74.6	3′-0″ ab fl @ A(FC)	33
Avg. Temp. 5′ (A,D,L,P)	74.40	3′-0″ ab fl @ D(FC)	39
Air Temp in Light Trough		**Metal Clg Pan Surf. Temp**	
Temp at ballast end	80.5	At location B	74.9
Temp at other end	76.65	At location C	75.25
Ceiling Space Temperatures		At location E	75.07
Bet. acoustic & pan @ J	74.9	At location F	74.8
Top of acoustic @ V	73.92	At location G	74.85
Center clg. space @ V	74.40	At location I	74.6
0′-2″ below slab @ V	74.98	At location J	74.85
Avg temp ctr clg space	74.44	At location N	71.4
Temperature Below Ceiling		**Concrete Slab Temps**	
0′-2″ below clg @ B	74.6	Avg clg top surf.	74.57
0′-2″ below clg @ C	74.77	Avg clg mid. plane	74.63
0′-2″ below clg @ E	74.36	Avg clg bot surf.	74.58
0′-2″ below clg @ F	74.27	Floor top surf. @ Y	74.72
0′-2″ below clg @ G	74.27	Floor mid. plane @ Y	74.67
0′-2″ below clg @ I	73.93	Floor bot surf. @ Y	73.55
0′-2″ below clg @ J	74.27	**Perimeter Slab Temp**	
0′-2″ below clg @ N	73.54	Clg mid pl @ W wall	71.1
Psychrometer Readings		Clg bot sf @ W wall	72.34
Test space DB	74.9	**Heat Flow Meter Data**	
Test space WB	56.3	At clg-Btu/hr-sq ft	−0.71
Outside air (Weath Bur)	41.0	Clg meter temperature	75.0
Variation During Test		At fl-Btu/hr-sq ft	0.78
Recorder-DB-F	1.0	Floor meter temp.	75.3
Recorder-RH-%	2.0	(Neg value is heat out)	

various data, including results from their own tests on a completely instrumented environmental room and stating a correction factor for panel size. Another element of importance at low temperature differences is the effect of forced convection. This topic was investigated by Parmelee and Huebscher[11], who reported their results in terms of forced convection coefficients added to natural convection coefficients to obtain *h* values for total convection transfer. To illustrate the vast

deviation possible with different formulae, 7 cases have been plotted in Fig. 4. Values at the lower temperature differences represent extrapolation.

The top line, which was adjusted for panel size and forced convection, is basically the data of Wilkes and Peterson[12], who pioneered in using large panels and small temperature differences, with air temperature measurements 2½ in. from the panels. Near the lower end of the band are the ASHAE data which have the advantage of

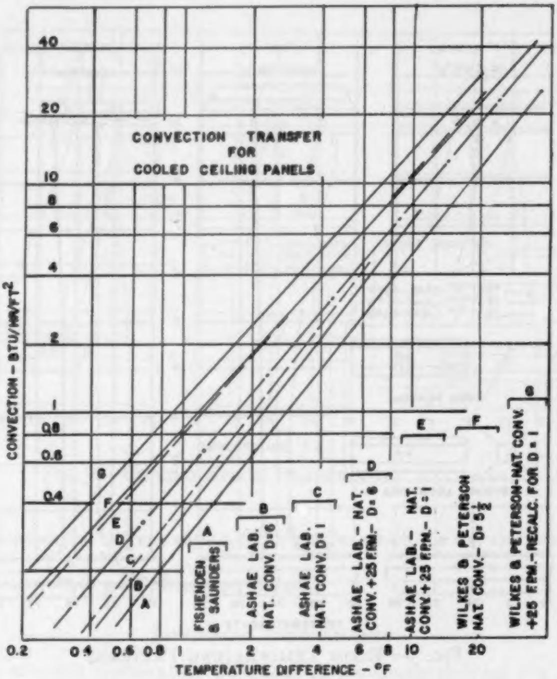


FIG. 4—CONVECTION FROM COOLED CEILING PANEL (LOWER VALUES EXTRAPOLATED)

relating to room size panels. They were calculated in terms of room temperature measured 5 ft above the floor. Actual cases show less difference between various formulae when values consistent with the respective test conditions are used. To calculate convection transfer for this experiment, the ASHAE formulae were applied for the surfaces above the ceiling using average ceiling space temperatures. The transfer for the sloping sides was evaluated by other data of Wilkes and Peterson for inclined surfaces[12]. Downward convection was computed by the Wilkes data using the local air temperature and taking into account the measured velocity at each area.

The quantity of heat transferred from other room surfaces to the water-cooled luminaire was calculated by conventional methods as presented by Raber and Hutchinson[18], who give various charts for accurate estimation of shape factors.

ROOM TEMPERATURE PATTERNS

Temperature patterns for 3 of the test runs are given in Figs. 5 and 6. At the *normal* condition, there was only a very small variation in temperature from floor

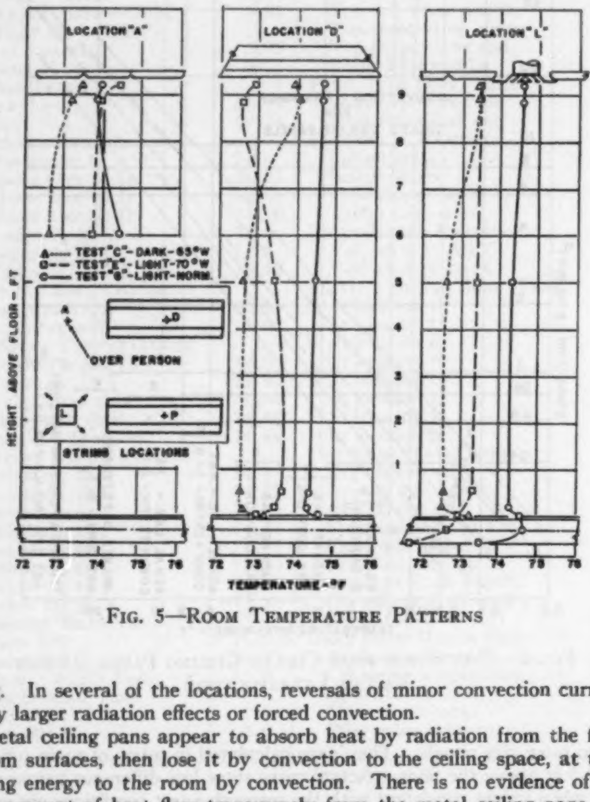


FIG. 5—ROOM TEMPERATURE PATTERNS

to ceiling. In several of the locations, reversals of minor convection currents are caused by larger radiation effects or forced convection.

The metal ceiling pans appear to absorb heat by radiation from the floor and other room surfaces, then lose it by convection to the ceiling space, at the same time losing energy to the room by convection. There is no evidence of any appreciable amount of heat flow transversely from the metal ceiling pans into the water-cooled panel extension of the luminaire.

EFFECTIVE TEMPERATURE DIFFERENCE

Because of the geometrically complex shape of the system, it is not practical to devise an empirical function to accurately relate several other variables to a true

effective temperature difference. However, examination of the data leads to a practical index. Table 2 gives an excerpt of the readings.

The mean of the average ceiling space temperature (measured midway between the ceiling and the slab above) and the average room temperature (at the 5-ft level) should be a suitable parameter of the roomside index for the following reasons:

1. It represents an average air temperature generally surrounding the luminaire.
2. It approaches (though it is not identical to) the average room temperature.

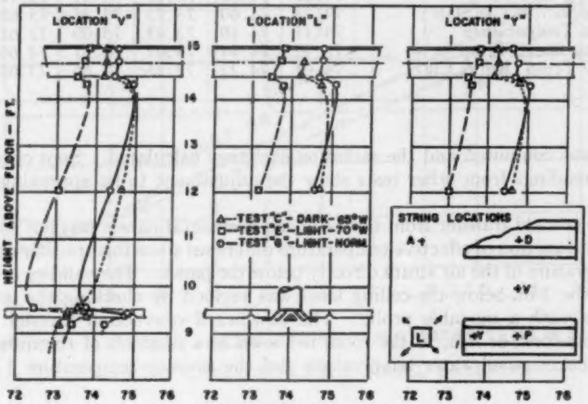


FIG. 6—CEILING SPACE TEMPERATURE PATTERNS

3. It approaches the average surface temperatures which the top and bottom of the luminaire *see*.
4. Its use resulted in good correlation of data.

LUMINAIRE PERFORMANCE

Since the effective temperature difference as previously defined is a significant parameter with respect to both radiation and convection, the 2 quantities may be added and shown on a single chart. For example, the heat transfer by radiation and convection from the top of the luminaire is shown as Fig. A-2 in the Appendix.

Total convection and radiation quantities for the various parts of the fixture are shown in the form of a complete heat balance in Table 3. Calculations for the radiometer readings were completed only for Test G for net exchange through the luminaire opening. This result is intended to be an order of magnitude appraisal rather than an accurate figure. Precise radiation measurements of this nature in other than laboratory surroundings are extremely difficult. Test G was the only one where the full series of measurements could be carefully made without interruption. No significant change in high temperature radiation would be anticipated for the other tests. In order to estimate the change in low temperature radiation, the area weighted average surface temperature of all surfaces visible through the

TABLE 2—EXCERPT OF LOGS TO DETERMINE EFFECTIVE TEMPERATURE DIFFERENCE

TEST NO.	F	G	E	D	C	K
CONDITION OF TEST	LIGHT			DARK		
Water Temp Ent Panels	68.83	62.27	58.90	69.27	63.92	55.28
Supply Air Temp @ Outlet	64.06	69.23	68.82	—	—	—
Avg Floor Surf. Temp	74.04	74.62	73.02	72.85	73.25	71.86
Avg Bot Clg Slab Temp	74.22	74.58	73.46	73.84	74.10	72.31
Avg of Flr Surf. & Clg Slab	74.13	74.60	73.23	73.35	73.63	72.08
Avg Room Temperature	74.11	74.40	73.43	73.00	72.62	70.34
Avg Ceiling Space Temp	73.92	74.44	73.07	74.31	74.66	72.42
Avg Space Temp (Rm & Clg)	74.01	74.22	73.25	73.66	73.64	71.38

opening was computed and the radiation exchange calculated. Spot checks of individual readings from other tests show the adjustment to be approximately correct.

The downward transfer from the extended horizontal panels was not assumed to be a direct function of effective temperature difference since this transfer depends on the temperature of the air strata directly below the panel. The validity of measurement at the 2-in.-below-the-ceiling level was verified by checking the gradient in the strata with a movable probe. The downward convection transfer from the panel extensions, or *lips*, to the room is shown as a function of the difference between average panel-water temperature and the average temperature 2 in. below

TABLE 3—HEAT BALANCE ON WATER-COOLED LUMINAIRE
(50 *sq ft floor area per fixture*)

TEST DESIGNATION	F	G	E
HEAT REJECTED BY FIXTURE	Btu per Hour		
1. Top Surfaces—Convection & Radiation	60.7	11.2	—
2. Center Pieces—Convection	11.5	15.0	17.2
3. Radiation fr. Opening to Room	62.0	50.0	43.0
4. Fluorescent Tube Convection	26.2	35.5	46.7
5. Heat Absorbed by Panel Water	199.0	370.5	420.0
TOTAL	359.4	482.2	526.9
HEAT ABSORBED BY FIXTURE			
1. Horizontal Panels—Convection & Radiation fr. Room	17.0	63.0	77.1
2. Top Surfaces—Convection & Radiation	—	—	15.3
3. Trough Faces—Roomside Convection	33.1	39.5	38.5
4. Electrical Energy	358.0	368.0	373.5
TOTAL	408.1	470.5	504.4
DIFFERENCE	48.7	11.7	22.5
% ERROR	10.1	2.4	4.5
AVG SPACE TEMP. MINUS AVG. WATER TEMP, — °F	3.06	8.64	10.37

the ceiling in Fig. 7. As the *dark* tests were run without supply air, the calculations for these tests do not include any forced convection component.

In this installation, there was a definite, though minor, difference between the temperature difference based on average space temperature and that based on 2 in. below the ceiling (the relation between the two can readily be plotted from the

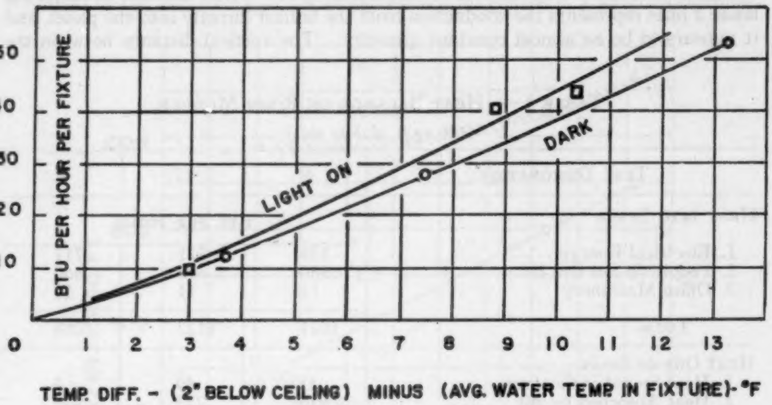


FIG. 7—DOWNWARD CONVECTION FROM PANEL EXTENSIONS (LIPS)

data). Greater variations might arise, depending on the quantity of air and type of distribution used.

STORAGE

The most consistent method for estimating heat input to the floor was found to be by the heat meter, which had a good bond to the floor surface. Heat flow into the overhead slab was estimated from readings of the buried thermocouples, as explained in the Appendix. The amount of flow into the slabs was small; the proximity of a person at times reversed the direction of flow. Such small magnitudes resulted in large percentage errors in these items.

An approximate energy accounting for a 100 sq ft area from slab to slab is presented in Table 4. This summary includes one air diffuser and 2 water-cooled luminaires. The heat gain from people was calculated from the average occupancy of 720 sq ft of area immediately surrounding the test fixture. The error in the balance is generally less than the value estimated for one person in the 720 sq ft. As expected, the amount of heat which can be assigned to storage, that is, the net difference in heat energy to the slabs, is extremely small. The relative pickup by the panel and the air is graphically represented in Fig. 8.

OVERALL PERFORMANCE

The conduction and independent transfer were adjusted for variation from design voltage. In assembling the data, the natural convection and room radiation curve

was computed from the 4 fixtures preceding the fully instrumented test panel, all operating dark. The test unit, with only the ballast operating, furnished the points for the middle curve. The *normal operation* data were taken from the average of 5 units, with the test unit shown separately. The test unit performance was slightly higher because it contained lamps which had operated a shorter time and because it was one of the fixtures in line with a diffuser. The overall performance is plotted in Fig. 9 which shows the flat curves expected. The vertical distance between the lower 2 lines represents the conduction from the ballast directly into the panel, and it appears to be an almost constant quantity. The vertical distance between the

TABLE 4 — HEAT BALANCE ON ROOM MODULE
(100 *sq ft, slab to slab*)

TEST DESIGNATION	F	G	E
HEAT INTO SPACE	BTU PER HOUR		
1. Electrical Energy	736	721	717
2. People @ 240 Btu Hr.	330	390	360
3. Office Machinery	11	11	11
TOTAL	1077	1122	1088
HEAT OUT OF SPACE			
1. Heat Stored in Concrete	51	59	6
2. Heat Absorbed by Air	513	253	226
3. Heat Absorbed by Panel Water	464	796	896
TOTAL	1028	1108	1128
DIFFERENCE	49	14	40
% ERROR	4.5	1.2	3.5
AVG SPACE TEMP MINUS AVG WATER TEMP —°F	4.01	10.22	12.15

top 2 curves is the quantity defined as *Independent Transfer*. The relatively constant height of this band confirms the virtually independent nature of this function. Extrapolating to the ordinate, it can be seen that if the panel were operated at room temperature, it would absorb 35 percent of the energy supplied to the lights. By operating at 9 to 10 deg ET (effective temperature) difference, more than 100 percent of the equivalent energy supplied to the lights will be transferred by the complete assembly to the cooling water. At greater temperature differences, even more heat will be transferred from the lights and room to the water.

The results indicate that the heat-absorbing paint improved the characteristics only slightly, probably because re-reflections in the mouth-shaped luminaire troffer produced an initially high effective absorption, even with the regular paint. The addition of a glass lens tends to slightly increase the amount of heat pick-up by the water and might be used where desirable for lighting effect.

There was some question as to how the performance in the interior zone would actually vary from summer to winter. Examination of strip chart records of permanent concrete couples showed identical temperatures in the center of the slab in

mid-August as in February when these tests were made. It had also been found that supply air and supply water temperatures to the interior zones had to be maintained substantially at the same control points summer and winter.

Conclusions

The basic instrumentation and methods appeared to be adequate to obtain a realistic indication of the heat exchange mechanisms which take place in a water-cooled lumi-

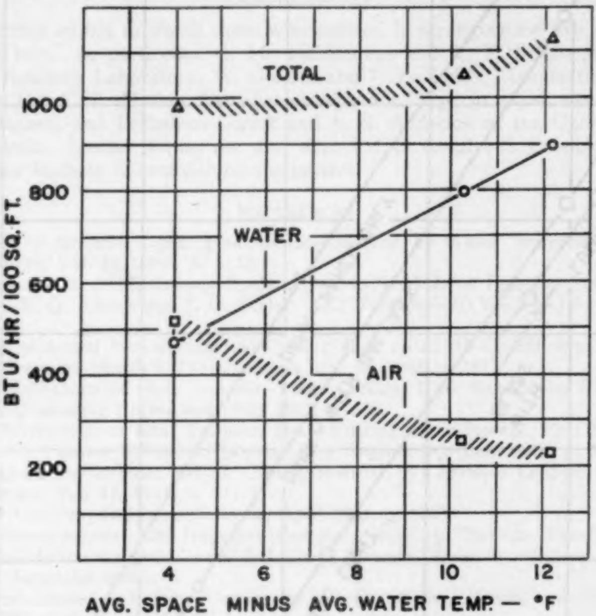


Fig. 8—Heat Pickup by Panels and Air

naire operating in a panel-air system. It was found that many of the transfer processes result from extremely small temperature differences, and what precision could have been gained in a laboratory might have been lost due to lack of similarity to the real application. Compared to the conventional type of system, it was found that radiation from the luminaire enclosure and heat absorbed by the structure were extremely small, attesting to the fact that the fixture is a good *radiation trap*.

The experiment indicates that no single location of room temperature is entirely valid as a variable regulating panel transfer. The effective temperature difference, defined as average space temperature minus average water temperature, was found to be a suitable index.

The overall performance curve is valid for the type of fixture tested. The additional curve in Fig. 7 shows the variation of the pick-up by the extended panels and might be used to adjust the overall performance curves for various ceiling layer temperatures found with different types of outlets and supply air temperatures.

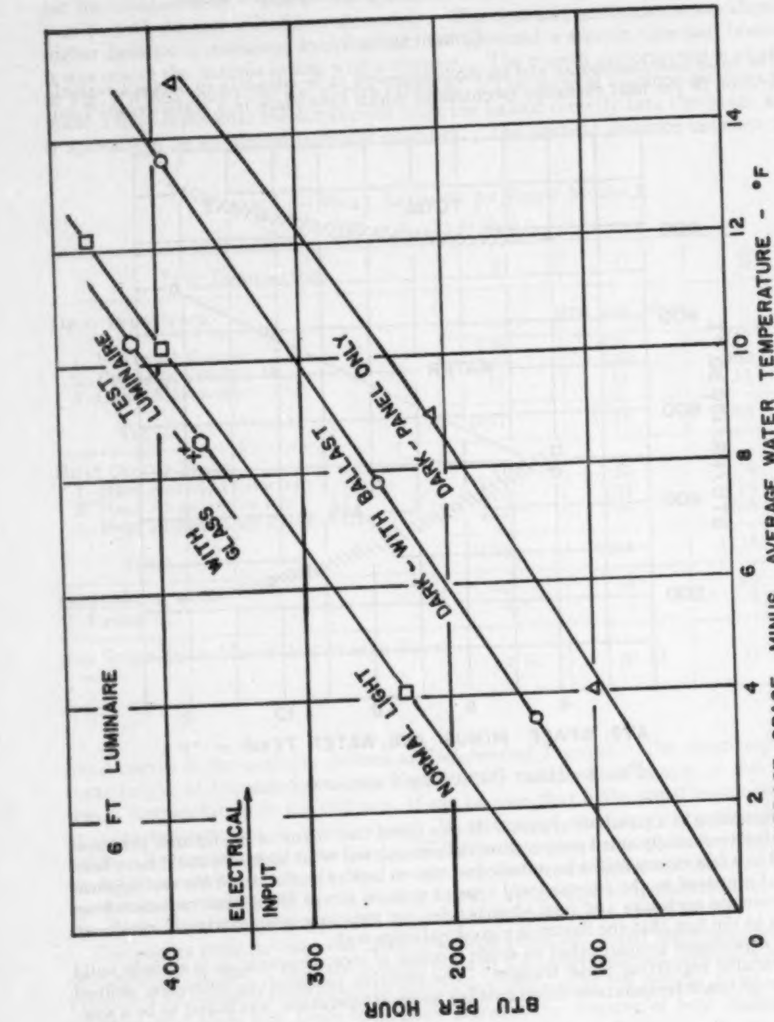


FIG. 9—OVERALL PERFORMANCE OF 6-FT WATER-COOLED LUMINAIRE

These data illustrate the type of heat transfer mechanisms that prevail in actual operation of a panel-air system and indicate the capability of specifically designed water-cooled luminaires to absorb the entire cooling load from lighting, and transfer additional heat from the room.

Recalling the original objective, it would be practical, for example, to cool a 200 sq ft section of interior floor area of a normal general office space using 2 lines of ½ in. copper tubing (supply and return) and 2 runs of 4½ in. round ducts conveying air at conventional velocities and pressure drop.

Acknowledgment

The author wishes to thank those who assisted in arranging for this field performance test. In particular, C. M. Humphreys and L. F. Schutrum, of the ASHAE Research Laboratory; W. Gordon and J. Y. Smith, Manufacturers Life Insurance Co.; C. L. Mulloy, Canadian Acceptance Corp., in whose area the test was conducted; and Professors Gulick and J. R. Anderson of the University of Pennsylvania. Special thanks are also extended to Charles S. Leopold, Philadelphia, for his help in establishing the project.

References

1. Effects of Artificial Lighting on Air Conditioning, by Walter Sturrock (ASHVE Transactions, Vol. 44, 1938, p. 213).

2. A New Means of Minimizing Radiant Heat for High Level Lighting Systems, by D. Cannel, E. Q. Adams and J. C. Forbes (*IES Transactions*, Vol. 34, July 1939, p. 726).

3. Study of Actual Versus Predicted Cooling Load on an Air-Conditioning System, by J. N. Livermore (ASHVE Transactions, Vol. 49, 1943, p. 287).

4. The Mechanism of Heat Transfer, Panel Cooling, Heat Storage, by Charles S. Leopold (*Refrigerating Engineering*, July 1947, p. 33).

5. The Mechanism of Heat Transfer, Panel Cooling, Heat Storage, Part II—Solar Radiation, by Charles S. Leopold (*Refrigerating Engineering*, June 1948, p. 571).

6. Design Factors in Panel and Air Cooling Systems, by Charles S. Leopold (ASHVE Transactions, Vol. 57, 1951, p. 61).

7. Panel Cooling (*Architectural Forum*, April 1951, p. 166).

8. The Manufacturers Life Insurance Company Building, Toronto; Panel Cooling for the Air Conditioning, by Charles S. Leopold (*Journal Royal Architectural Institute of Canada*, November 1953).

9. Natural Convection in Panel Heating, by J. R. Carroll, Jr., (*Heating and Ventilating*, January 1948, Vol. 45, pp. 70–76).

10. ASHAE Research Report No. 1576—Natural Convection and Radiation in a Panel-Heated Room, by T. C. Min, L. F. Schutrum, G. V. Parmelee and J. D. Vouris (ASHAE Transactions, Vol. 62, 1956, p. 337).

11. ASHVE Research Report No. 1316—Forced Convection Heat Transfer From Flat Surfaces, by G. V. Parmelee and R. G. Huebscher (ASHVE Transactions, Vol. 53, 1947, p. 245).

12. Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (ASHVE Transactions, Vol. 44, 1938, p. 513).

13. *Panel Heating and Cooling Analysis*, by B. F. Raber and F. W. Hutchinson (John Wiley & Sons, New York, N. Y., 1947).

APPENDIX

Test Set-Up and Instrumentation

The most important readings were the water temperatures. Special thermometer wells, designed to reflect water temperatures to within 1/100 F deg of the stream tem-

perature were constructed and inserted at the inlet and outlet of the string of 5 panel fixtures and at the inlet of the fifth fixture which had been selected for more detailed instrumentation. A calibrated orifice and mercury manometer were installed for flow measurement, with an auxiliary pump and throttling valve to adjust flow. Differential thermopiles were used to measure water temperature differences across the panels. A millivolt amplifier was used to multiply the output of the thermopiles by 100.

Fig. 2 shows a part ceiling plan of the area surrounding the instrumented test fixture. The heavy dotted lines represent the suspended perforated metal ceiling pans which measure 1 ft x 2 ft and have a paper bound, glass fiber pad in the top, resting on a wire spacer between the pan and pad. At locations D, L, and P, 30-gauge thermocouples were attached at the floor surface 2 in. above the floor, 6 in. above the floor, at 5 ft, at 6 in. below the ceiling, and 2 in. below the ceiling. At location A, above a stenographer's desk, couples were installed 6 ft above the floor and 2 in. and 6 in. below the ceiling. Additional couples were installed 2 in. below the ceiling at locations B, C, E, F, G, H, I, J, M, and N. Surface couples were soldered to the metal pans immediately above these points. A string of couples was installed in the ceiling space at location V, with couples at the top of the acoustic pad, 2 in. above the pad, midway in the ceiling space, and 2 in. below the slab above. Additional ceiling space temperatures were taken at locations L, Y, and P and on top of the acoustic pads at locations H and N.

Two separate methods were used for estimating the flow of heat at the floor and ceiling concrete slabs. First, thermocouples were buried in the mid-plane of the slabs and attached to the surfaces at locations I, K, Y, and U at the ceiling and location Y at the floor. In addition, 3 concrete temperature couples and one room temperature couple were installed at the center of a private office adjacent to the west exterior wall. Considering the transfer of heat at the surface of a slab, it can be shown that if an infinitely wide slab of normal thickness is plunged into a medium of different temperature, the difference between the surface and mid-plane temperature can provide a reasonable estimate of the heat transfer at the surface over a considerable range of elapsed time, initial temperature difference, and surface heat transfer coefficient. In this way, the thermocouple readings were used to calculate an estimate of the transfer to the slabs. For the other method, 4 in. commercial bakelite heat flow meters were attached to the floor and ceiling slab surfaces in the location shown in Fig. 2.

Temperature of the supply air was measured at the outlet, location L, and at a trunk duct off the supply riser, approximately 25 ft upstream of the outlet. In each case, 4 couples were electrically averaged. Couples were attached to the water cooled luminaire, as shown in Fig. 3. Only one-half of the fixture was instrumented and temperatures for the other half were calculated. In the space surrounding the fluorescent tubes several 36-gauge couples were installed.

Copper constantan thermocouples were selected with minimum circuit resistance to obtain desired readability with a portable precision potentiometer. Concrete temperature couples were 20 gauge with output amplified ten times by a commercial millivolt amplifier, resulting in a readability of approximately ±0.05 F deg. Air temperature couples were 30 gauge with 24-gauge leads of similar calibration, resulting in readability of approximately 0.15 F deg. These couples were practically immune to radiation up to approximately 3 in. from the fluorescent tubes when the axes were not parallel. 36-gauge butt-welded couples were virtually immune to within 1½ in. from the tubes and were used with multiple strand constantan leads.

Following the field tests, each type of couple wire was carefully re-calibrated through the range of temperatures read during the tests.

Fixture surface temperatures were measured by couples which were aluminum soldered to the surface with 2 in. of the leads taped to the panel. Couples were spaced from the tubes to positions approximating the average surface temperature between tubes, with the assumption that the distance to the edge of the fixture was equivalent to half the tube spacing. Temperature gradients between tubes were calculated by a modification of the thin rod formulae.

Electrical measurements were made with portable meters which were later calibrated. Leads from the test fixture were brought down to an accessible switching arrangement.

Before any readings were taken for the tests, the *fifth* fixture was re-lamped with tubes which were known to have been operating for a period of from 4 to 8 weeks, since it is known that fluorescent tubes decrease in output rather sharply during the first 600 to 800 hours of operation, and then show a rather steady, slow decline. The other 4 fixtures were tested in the *as found* condition. The lamps were commercial 72T12 standard cool white used with a 59G773 ballast. A curve showing the spectral distribution of radiant energy for this type of lamp is shown in Fig. A-1, using data supplied by the lamp manufacturer.

Foot candle readings were taken at locations A and P with a commercial photographic type meter with uncorrected incident attachment. Subsequent comparison of the

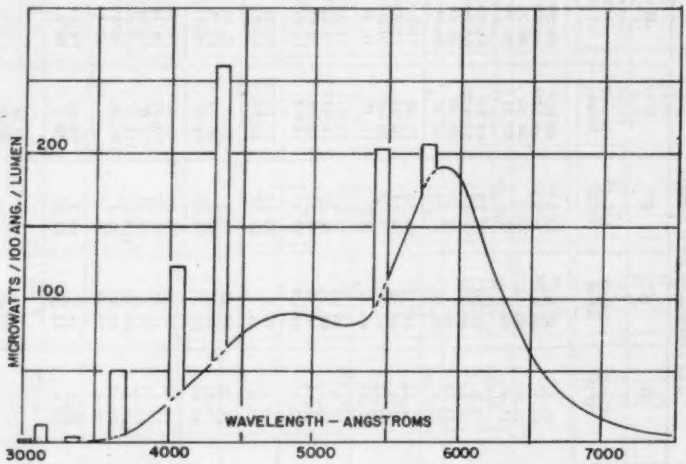


FIG. A-1—SPECTRAL DISTRIBUTION OF STANDARD COOL WHITE LAMP

meter used with a commercial Model 614 with visual illumination correction filter showed that under the light of the standard cool white lamp, the photographic meter read slightly low.

Air quantities were measured with a pitot tube and draft gauge at the trunk duct. The proportion of air through each of the 25 outlets surrounding the test area was determined by a commercial air meter with a specially brazed tip to fit over the lip of the diffuser.

The radiometer was a special instrument obtained from the ASHAE Laboratory. It has been previously described by Parmelee and Schutrum*. The instrument is normally used in a laboratory with controlled temperature bath, but approximate readings can be obtained in the field in the absence of strong radiation if the instrument is handled carefully. It was used with the output amplified 200 times. The calibration

* ASHAE RESEARCH REPORT No. 1528—Measurement of Angular Emissivity, by Aydin Umur, G. V. Parmelee and L. F. Schutrum (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 111).

TABLE A-1—SUMMARY OF TEST READINGS

(All temperatures are F degrees)

TEST NUMBER	C	D	E	F	G	H	I	K	L
	DARK 65°W	DARK 70°W	LIGHT 60°W	LIGHT 70°W	LIGHT NORM.	GLASS IN FIXTURE		HA DARK	HA LIGHT
LIGHT FIXTURE TEMPS									
Outer lip-free end	68.53	71.3	67.02	72.5	69.62	68.89	69.0	62.33	68.30
-free mid	68.41	71.3	66.6	72.5	69.23	68.50	68.74	62.15	68.08
-bal. mid	68.34	71.6	66.53	72.5	69.16	68.32	68.74	61.7	67.86
-bal. end	68.34	71.4	66.6	72.5	69.11	68.30	68.42	61.97	67.92
Inner lip-free end	68.5	71.4	67.80	72.72	69.85	69.10	69.30	62.62	68.57
-free mid	68.34	71.4	66.7	72.63	69.47	68.70	69.03	62.20	68.22
-bal. mid	68.7	71.73	66.7	72.72	69.40	68.80	69.22	62.15	68.23
-bal. end	68.51	71.6	66.92	72.59	69.40	68.80	69.11	62.33	68.28
Bot. side-free end	69.08	71.65	68.82	74.52	71.40	71.18	71.73	63.68	70.38
-free mid	68.63	71.4	67.80	74.13	70.14	71.82	71.4	62.42	69.03
-bal. mid	69.45	72.47	67.70	73.53	70.30	70.0	70.12	62.70	69.20
-bal. end	69.13	71.88	68.12	74.9	71.18	70.42	70.62	63.80	69.55
Top side-free end	69.08	71.6	70.03	75.2	72.17	72.25	72.47	63.60	71.18
-free mid	68.8	71.7	68.75	74.62	71.43	71.18	71.17	62.80	70.22
-bal. mid	71.6	74.73	70.35	76.52	73.15	72.68	71.0	64.80	71.82
-bal. end	69.4	72.3	69.05	75.3	71.73	71.60	72.93	63.52	70.48
Top ctr pc-free end	70.38	72.2	76.1	80.43	77.97	79.00	79.0	65.78	76.92
-bal. end	80.58	82.5	84.4	88.35	86.77	87.32	87.58	76.0	112.9
Bot ctr pc-free end	72.0	72.93	75.63	78.4	77.3	77.90	78.1	68.52	75.92
-middle	76.0	76.97	78.97	80.7	80.73	81.20	80.42	72.58	79.21
-bal. end	91.66	92.65	92.52	94.16	94.6	96.27	96.62	87.40	92.76
Top sheet-free end	71.4	72.68	73.97	76.85	75.33	75.51	75.73	67.50	74.10
-free mid	71.43	73.1	72.85	76.17	74.77	74.22	74.44	66.65	73.32
-free ctr.	74.43	76.21	75.3	78.53	77.07	76.86	77.82	69.67	75.73
-bal. ctr.	111.6	113.3	108.46	110.6	111.2	110.6	67.25	105.55	108.95
-bal. mid	95.72	97.22	93.42	96.3	96.06	95.4	95.83	89.73	93.98
-bal. end	74.89	76.5	76.38	79.45	—	78.3	—	70.74	74.5
Air in trough-free end	73.05	73.1	75.3	78.2	76.65	75.35	75.20	72.5	75.9
-bal. end	75.1	76.8	75.9	83.0	80.5	79.4	79.10	—	—
FLUO TUBE SURF TEMP									
West tube-free end	—	—	—	—	87.0	—	—	—	91.5
-bal. end	—	—	—	—	88.0	—	—	—	89.0
East tube-free end	—	—	—	—	87.0	—	—	—	89.5
-bal. end	—	—	—	—	90.0	—	—	—	87.0

Electrical Readings									
Ballast volts	130	130	125	121.5	123	122	123	—	121
amps.	—	—	—	—	—	—	0.898	—	—
watts	113.4	115.4	109.3	105	107.8	105.4	105.7	113.4	105.7
Tube amps. west	0.242	0.245	0.229	—	—	0.225	—	0.223	0.205
east	—	—	—	—	—	—	—	0.251	0.220
Tube volts west	181	184	—	—	185	184	181	—	188
east	186	189	—	—	185	186	185	—	195
Air Vel. Under Glass									
S.W. corner	—	—	—	—	—	10–60	—	—	—
S.E. corner	—	—	—	—	—	10–25	—	—	—
N.W. corner	—	—	—	—	—	5–15	—	—	—
N.E. corner	—	—	—	—	—	5–10	—	—	—
Glass Surf. Temp.									
Average of 4 readings	—	—	—	—	—	75.25	—	—	—
Light Meter Reading									
3 ft. ab. flr. at A	—	—	—	—	33 FC	29 FC	—	—	—
3 ft. ab. flr. at D	—	—	—	—	39 FC	38 FC	—	—	—
Temp Below Ceiling									
2" bel. clg. at B	73.35	73.67	73.77	73.93	74.6	73.11	73.88	70.4	73.06
at C	73.53	73.52	73.8	73.97	74.77	73.75	74.3	70.48	73.76
at E	73.53	—	—	—	74.36	75.54	73.84	70.30	72.92
at F	73.16	73.52	73.33	72.98	74.27	73.11	73.5	70.03	73.17
at G	73.12	73.45	73.71	73.83	74.27	73.75	74.37	70.48	73.4
at H	73.87	73.88	—	68.11	—	—	—	—	71.35
at I	73.37	73.5	72.68	73.4	73.93	73.02	74.15	—	72.68
at J	73.1	73.45	73.4	73.13	74.27	73.40	73.92	69.92	72.92
at M	73.58	—	71.35	—	—	68.96	69.72	—	70.80
at N	73.58	73.67	72.6	72.12	73.54	71.56	72.42	70.03	71.93
Temp. Bet. Acous. & Pan									
At location J	73.28	73.37	73.87	75.12	74.90	73.84	74.3	71.0	73.41
H	73.46	73.46	73.77	74.61	74.6	73.53	73.97	71.0	73.10
M	73.75	73.8	71.22	68.68	71.73	70.43	70.8	69.40	70.41
Supply Air Temps.									
At trunk duct	—	—	67.80	61.22	67.98	66.23	66.7	—	66.76
At outlet	—	—	68.82	64.06	69.23	67.76	68.11	—	67.92
Metal Clg Pan Temps									
At location B	73.38	73.28	73.8	74.4	74.9	74.30	74.77	71.3	73.28
C	73.38	73.28	74.0	74.76	75.25	74.78	75.08	71.22	73.53
E	73.53	73.4	73.58	73.96	75.07	74.30	74.3	71.22	73.28
F	73.32	73.75	73.58	74.27	74.8	74.33	74.4	70.97	73.23
G	73.14	73.15	73.54	70.0	74.85	74.40	74.4	70.97	73.20
H	73.53	74.58	73.37	72.67	76.42	73.14	73.5	70.88	70.30
I	73.46	73.4	73.8	73.96	74.6	73.97	74.19	—	69.50
J	73.26	73.32	73.75	70.5	74.85	74.22	74.4	70.97	73.40
N	73.85	73.7	72.2	71.4	73.46	72.6	74.4	70.41	73.32
M	73.76	74.03	70.57	67.68	71.4	70.42	70.12	69.21	69.91

TABLE A-1 (Continued)—SUMMARY OF TEST READINGS

(All temperatures are F degrees)

TEST NUMBER	C	D	E	F	G	H	I	K	L
	DARK 65°W	DARK 70°W	LIGHT 60°W	LIGHT 70°W	LIGHT NORM.	GLASS IN FIXTURE		HA DARK	HA LIGHT
AIR VELOCITIES, FPM									
At flr. at loc. D	0	—	0	—	—	—	—	—	8–12
J	0	—	0	—	—	—	—	—	—
P	0	—	—	—	—	—	—	—	—
5 ft. ab. flr. at L	15–50	—	10–45	—	—	—	—	—	—
2" bel. clg. at A	—	—	5–10	12	5	—	—	—	10–15
at B	—	—	10–30	8	—	—	—	—	—
at C	—	—	5–50	12	5	—	—	—	—
at D	—	—	—	10	—	—	—	—	0–6
at E	—	—	10–35	25	5–40	—	—	—	—
at F	—	—	10–35	8	5–10	—	—	—	13–23
at G	—	—	0–5	8	—	—	—	—	12–15
at H	—	—	20–40	85	50–70	—	—	—	15–78
at I	5–15	—	5–23	55	—	—	—	—	5–22
at J	—	—	15–25	15	0	—	—	—	10–25
at M	—	—	170	225	200	—	—	—	170–215
at N	—	—	40–100	110	110	—	—	—	75–120
at P	—	—	—	20	—	—	—	—	0–9
OCCUPANCY									
Acc. Mach. Operators	0	0	1	—	1	1	—	0	1
Add. Mach. Operators	0	0	5	—	5	5	—	0	4.5
Other Occupants	1	1	6	—	7	5	—	1	2
ROOM TEMPERATURES									
6'-0" above floor at A	72.63	73.1	73.71	74.23	74.36	73.67	74.2	70.40	73.32
0'-6" below clg. at A	73.16	75.53	73.88	74.23	73.84	73.14	73.75	70.48	73.11
0'-2" at A	73.46	73.53	74.5	75.28	73.92	73.32	73.75	70.48	73.71
Floor surface at D	73.2	72.68	73.12	—	74.6	—	74.1	71.82	72.84
0'-2" above floor at D	72.54	72.8	73.5	74.13	74.27	73.4	74.14	70.60	72.68
0'-6" at D	72.6	72.54	73.63	74.61	74.36	73.14	73.92	70.40	72.84
5'-0" at D	72.6	72.6	73.47	74.32	74.36	73.58	—	70.30	73.46
0'-6" below clg. at D	74.05	73.65	72.6	74.1	74.6	74.35	74.45	71.20	74.45
0'-2" at D	73.9	73.6	72.9	74.35	74.6	74.55	74.8	70.70	74.45
Floor surface at L	73.28	73.0	73.0	73.46	74.57	—	73.92	71.82	72.2
0'-2" above floor at L	72.54	72.83	73.28	73.93	74.27	73.5	73.9	70.48	73.11
0'-6" at L	72.63	72.8	73.32	73.93	74.57	73.5	73.9	70.33	73.14
5'-0" at L	72.63	73.1	73.42	73.93	74.3	73.8	74.2	70.33	73.41
0'-6" below clg. at L	73.46	73.5	73.47	73.41	74.53	73.85	74.14	70.79	73.06
0'-2" at L	73.4	73.84	73.47	73.5	74.53	74.18	74.23	70.52	73.28

Floor surface at P	73.16	72.83	72.85	74.37	74.6	73.5	74.1	71.95	72.2
0′-2″ above floor at P	72.68	72.83	73.03	73.93	74.27	73.54	73.97	71.13	72.54
0′-6″ at P	72.63	72.8	73.12	73.83	74.14	73.46	73.84	70.79	73.02
5′-0″ at P	72.63	73.1	73.12	73.97	74.57	73.54	73.9	70.33	73.11
0′-6″ below clg. at P	73.35	73.5	73.6	75.4	74.9	75.55	74.7	70.95	74.05
0′-2″ at P	73.35	73.85	73.0	73.95	74.6	73.5	74.3	71.0	74.30
Avg. room temp. at 5 ft.	72.62	73.0	73.43	74.11	74.40	73.65	74.1	70.34	73.32
PERIMETER ROOM TEMP.									
5′-0″ above floor at Q	72.01	72.83	72.90	74.42	73.22	72.43	73.95	69.76	—
CEILING SPACE TEMPS.									
Top of acoustic at V	73.58	73.72	72.47	72.9	73.92	72 97	73.03	71.26	72.26
0′-2″ above clg. at V	—	—	—	—	—	—	—	70.30	71.56
Center clg. space at V	74.7	74.7	73.03	74.13	74.40	73.76	73.75	72.6	72.93
0′-2″ below slab at V	74.83	74.83	73.75	74.52	74.98	74.37	74.5	73.02	73.5
Top of acoustic at B	73.71	73.76	72.47	73.0	73.53	72.63	72.72	70.79	72.17
Top of acoustic at H	73.71	73.88	72.47	72.84	73.76	72.9	73.03	71.30	72.17
Bet. acoustic & pan at H	73.46	73.46	73.77	74.61	74.60	73.53	73.97	71.0	73.1
at J	73.28	73.37	73.87	75.12	74.90	73.84	74.3	71.0	73.41
at M	73.75	73.80	71.22	75.2	72.17	70.43	70.8	69.4	70.41
Center clg. space at L	74.67	74 7	73.12	74.1	74.60	74.37	73.92	72.63	72.93
0′-2″ below slab at L	74.7	74.65	73.75	74.4	75.10	—	74.37	73.02	73.41
Center clg. space at Y	74.6	72.77	73.0	73.5	74.40	74.22	73.75	72.08	72.63
0′-2″ below clg. at Y	74.77	74.65	73.53	74.4	75.02	73.83	74.37	73.05	73.41
Top of acoustic at N	73.71	73.88	72.47	72.25	73.72	72.47	72.83	71.26	72.03
Center clg. space at P	74.67	75.07	73.15	73.93	74.36	73.67	73.63	72.53	72.68
0′-2″ bel. slab at P	74.67	74.6	73.93	74.4	75.06	74.14	74.4	73.11	73.32
Avg. center clg. space	74.66	74.31	73.07	73.92	74.44	74.01	73.76	72.42	72.79
PSYCHROMETER READINGS									
Test space D B	—	—	—	74.2	74.9	74.3	74.7	—	73.7
Test space W B	—	—	—	57.0	56.3	57.1	57.1	—	56.8
Outside (Weather Bureau)	27	31	33	38	41	36	37	31	41
VARIATION DURING TEST									
Recorder—Dry Bulb	—	—	¾	½	1	¼	—	1	½
—Rel. Hum.	—	—	½	1	2	1	—	1	2
PANEL WATER TEMPS									
Ent string 5 panels	63.92	69.27	58.90	68.83	62.26	61.60	61.76	55.28	61.31
Ent No. 5 test panel	65.61	70.03	62.46	70.73	65.43	64.88	65.09	57.53	64.89
Lvg string 5 panels	66.12	70.32	63.31	71.18	,66.14	65.79	66.06	58.02	64.45
PANEL WATER FLOW-GPM	0.985	0.965	1.06	1.03	1.01	0.978	1.02	1.385	1.015

TABLE A-1 (Continued)—SUMMARY OF TEST READINGS

(All temperatures are F degrees)

TEST NUMBER	C	D	E	F	G	H	I	K	L
	DARK 65°W	DARK 70°W	LIGHT 60°W	LIGHT 70°W	LIGHT NORM.	GLASS IN FIXTURE		HA DARK	HA LIGHT
WATER TEMP DIFF (PILE)									
String of 5 panels	2.164	1.108	4.233	2.250	3.948	4.157	3.848	2.780	3.75
Test panel No. 5	0.504	0.301	0.791	0.387	0.733	0.786	0.743	0.568	0.691
CONCRETE SLAB TEMPS									
Ceiling top surf. at K	74.23	73.97	73.82	74.45	74.60	74.11	74.13	73.13	72.83
mid plane at K	74.28	74.02	73.74	74.42	74.53	74.02	74.09	73.67	72.84
bot surf. at K	74.12	73.87	73.49	74.25	74.44	73.88	74.0	72.38	72.77
mid plane at I	74.14	73.70	73.77	74.50	74.75	74.22	74.27	68.90	72.86
bot surf. at I	74.03	73.83	73.48	74.32	74.72	73.87	74.03	72.32	72.78
top surf. at U	74.22	73.91	73.77	74.38	74.53	74.03	74.13	73.06	72.88
mid plane at U	74.23	73.93	73.69	74.29	74.53	74.09	74.17	72.97	72.82
bot surf. at U	74.14	73.83	74.43	74.10	74.53	73.92	73.98	71.73	72.78
mid plane at Y	—	—	73.7	74.38	74.73	74.14	74.32	71.68	72.62
bot surf. at Y	—	—	73.43	74.20	74.63	73.93	74.1	72.80	72.62
Floor top surf. at Y	73.36	72.9	73.1	74.29	74.72	74.43	74.12	72.02	72.34
mid plane at Y	73.41	72.92	72.88	74.21	74.67	74.40	74.0	71.50	72.07
bot surf. at Y	—	—	71.93	73.52	73.55	73.48	73.20	71.82	71.0
Wall surf. at W	73.31	73.38	73.27	73.90	74.52	73.90	73.83	70.95	73.02
Ceiling beam surf.	—	—	73.54	74.22	74.66	73.97	73.97	72.60	72.68
Average clg. top surf.	74.22	73.94	73.80	74.42	74.57	74.07	74.13	73.09	72.86
clg. mid plane	74.22	73.88	73.72	74.40	74.63	74.22	74.21	71.81	72.79
clg. bot surf.	74.10	73.84	73.46	74.22	74.58	73.90	74.03	72.31	72.74
(clg. top minus mid)	0	0.06	0.08	0.02	−0.06	−0.15	−0.08	1.28	0.07
(clg. bot minus mid)	−0.12	−0.04	−0.26	−0.18	−0.07	−0.32	−0.18	0.50	−0.05
Floor (top minus mid)	−0.05	−0.02	0.22	0.08	0.05	0.03	0.12	0.52	0.27
Floor (bot minus mid)	—	—	−0.95	−0.69	−1.12	−0.92	−0.80	0.32	−1.07
PERIMETER SLAB TEMP.									
Ceiling mid plane at Q	71.5	71.24	—	71.25	71.1	70.70	70.95	69.5	69.92
bot surf at Q	71.83	71.68	—	71.50	72.34	70.76	71.02	69.79	70.18
HEAT FLOW METER DATA									
Heat flow at ceiling Btu/(hr)(ft²)	−0.57[a]	−0.116[a]	−0.864[a]	−0.992[a]	−0.706[a]	−0.717[a]	−0.83[a]	−2.145[a]	−0.452[a]
Meter temp at ceiling F	74.5	74.3	74.0	74.6	75.0	74.05	74.4	72.7	73.2
Heat flow at floor Btu/(hr)(ft²)	−0.61[a]	−0.1277[a]	+1.033	+0.536	+0.781	+0.127	+0.625	−2.31[a]	+1.530
HIGH TEMP RADIATION									
Ht. meter at flr. Btu/(hr)	—	—	—	0.405	0.360	—	0.160	—	—

[a] Negative value denotes heat flow out of slab surface

of the radiometer was checked by sighting into a high temperature heat treating oven whose temperature was measured by a thermocouple. The total radiation was computed by a modification of the general method described by Sturrock and Staley**, except that high and low temperature radiation were identified and adjustments made for change of distance of the instrument from the lamps.

Radiometer positions were set at 15 deg intervals at 9 locations around the center of the fixture and 3 around the end. At each of the locations the position of the radiometer

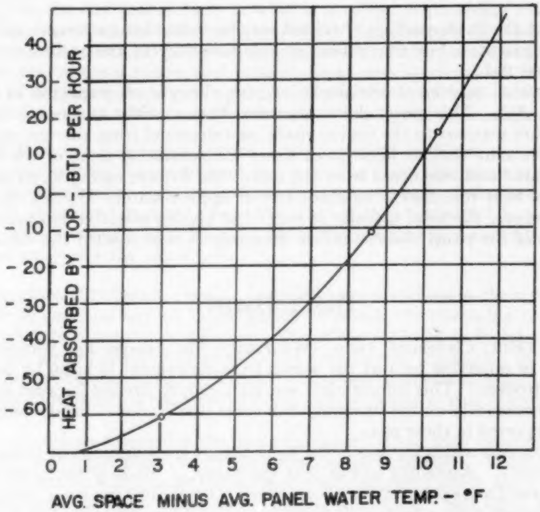


FIG. A-2—RADIATION AND CONVECTION FROM TOP OF FIXTURE

was such that the angle of the instrument would intercept the opening of the luminaire with a given angle to horizontal.

Radiometer readings were taken pointing the instrument at the fixture with shutter closed, opening it, repeating with the lights off, then with the shutter closed. Immediately following these readings, shutter and barrel temperatures were recorded. At each position a similar set of readings was taken with the radiometer pointing 180 deg from the luminaire. In this reverse position, no difference was obtained with lights on or off and the readings with lights off were omitted.

For each test the automatic controls of the air apparatus were adjusted to maintain approximately 74 F room temperature. At times, the apparatus was placed under complete manual control. Periodic readings were taken at the test area to evaluate the stability of conditions. A temperature recorder was used on the twelfth floor to monitor air temperatures in various parts of the supply apparatus. It was found that when the

** Fundamentals of Light and Lighting, by Walter Sturrock and K. A. Staley (*Bulletin LD-2*, Large Lamp Dept., General Electric Co., January 1956).

control system was stabilized by 10:30 a.m., the conditions in the test area would reach a nearly steady state by approximately 1:30 p.m. When check readings showed stability in the test area, logs were taken. Three readings were made of the water and air temperatures throughout the test and one reading was taken of the other quantities. Immediately after semi-darkness set in, tests were made with the floor heat meter to estimate the amount of short wave radiation incident on the floor. The heat flow meters came to equilibrium quickly, although subject to the proximity of people and small air currents. Having established a reading, all the lights in the room were quickly switched off and another reading taken before the fluorescent tubes cooled appreciably. The difference was an approximation of the short wave radiation incident upon the meter surface.

It is felt that the basic readings obtained may be valuable as reference for future design or investigations. For this reason, a full summary of the resultant readings is shown in Table A-1.

One curve which may be of additional interest, though not presented in the text, is shown in Fig. A-2. This curve shows the total heat transfer at the top faces of the fixture which are exposed to the ceiling space, as calculated from the test readings. It is interesting to note that at high panel water temperatures, some of the heat of the ballast is radiated and convected from the top of the fixture; as the water temperature is lowered, the heat rejection is reduced, and at approximately 9½ deg effective temperature difference, the total transfer is zero. At greater effective temperature differences, the top of the panel absorbs rather than rejects heat toward the ceiling space.

DISCUSSION

L. F. SCHUTRUM, Cleveland, Ohio, (WRITTEN): Mr. Spiegel has demonstrated the practicability of removing heat at the source from fluorescent lighting by means of the water-cooled troffers. This information was particularly needed because energy input to lighting is constantly on the increase, with some installations already using ten times the wattage reported in these tests.

One of the factors which influences the light output from fluorescent tubes is the bulb wall temperatures. According to Fig. 4 of the paper, *A Study of the Extraction of Heat from Fluorescent Luminaires in Air Cooled Rooms*, by Walter Sturrock, maximum light output is obtained with a bulb wall temperature of approximately 100 F. At higher and lower temperatures, the light output is reduced. Mr. Spiegel reports that the bulb wall temperatures are about 88 F in Test G, which means that the fluorescent tubes are performing at nearly maximum efficiency. However, if the water temperature is lowered much beyond the value given in this test, the lighting efficiency will begin to drop off rapidly.

An attempt was made to compare the performance of the water-cooled luminaire and the performance of the ASHAE Environment Laboratory in removing heat from suspended fluorescent lighting by means of a cooled ceiling panel in combination with an air system. The cooled ceiling panel, in this case, covered 100 percent of the ceiling area. Calculation showed that the ceiling panel would remove the same amount of heat as that given in Fig. 9 for the 6-ft water-cooled luminaire, if the temperature difference between the room air at the 60-in. level and the panel water were 2 F deg to 2½ F deg greater than for the water-cooled luminaire. The lighting, people, and machinery loads given in Table 4 were used for the calculations, and in addition, a conditioned-air rate of 0.85 cfm per sq ft of floor area, and a panel resistance of unity between the panel surface and the water in the tubes were assumed. No heat gain from above the panel was included.

The performance curve thus calculated would lie parallel to the Normal Light Curve of Fig. 9, but about 2 F deg to the right. When, however, a panel resistance of less than unity was used, the slope of the curve was steeper and crossed the curve for the normal light. This may indicate that the slope of the curves in Fig. 9 are also dependent upon

the thermal resistance between the panel water and the average panel surface. The lower the panel resistance, the steeper will be the curve and vice-versa.

One may speculate also that the performance of a panel system with recessed lighting without water-cooled luminaires would be better than for the suspended lighting. Research is needed on this matter.

The values of the heat absorbed by the water in Table 4 agree with those in Fig. 8, however, the heat absorbed by the air and the total do not agree. Would the author please explain what factors the discusser has overlooked in making this comparison?

H. E. ZIEL, Detroit, Mich.: This is a remarkable paper. I would like to inquire from the author if he has any cost data as to what is involved in providing this extra circuit for the cooling *vs* putting the same cooling effect in the air supply for ventilation.

I think that with some lights, there may be a condition where much of the light energy goes into the ceiling space above, and in many cases that space is used for recirculating air. If not in the paper already, it would be helpful to have the total heat input of the light divided into what comes into the room and what goes up into the ceiling space.

R. W. MCKINLEY, Pittsburgh, Penna.: My question is: Have you considered the possibility (after making the investment in the cooling coils) that these same coils might be used for heating? The lighting levels which are currently being installed in some locations, and which are reported to be in the range of standardization in the near future, will call for high cooling capacities. It would appear that the heating requirements could be met easily by the same coils.

E. S. HOWARTH, Pittsburgh, Penna.: I would like to compliment the author on a very excellent job in carrying out an investigation in which the obtaining of the required type of precise data is extremely difficult. The Alcoa Building in Pittsburgh, to which he referred, has water-cooled luminaires. The construction employed in that building consisted of attaching the specially designed aluminum fluorescent fixtures to the same water-carrying pipes which serve the aluminum ceiling panels in this heating and cooling installation.

I have two questions concerning the author's paper. One, with reference to the percentage of electrical input absorbed by the water-cooled luminaire. Is any undue credit given to the extensions which lie in the plane of the ceiling and which are in a position to receive energy from the room as an active ceiling would do? Secondly, and this may appear in the Appendix to which the author referred, what maximum temperatures were observed on the luminaire ballast?

AUTHOR'S CLOSURE: Mr. Zeil asked whether any cost data is available for the installation of a water-cooled luminaire circuit compared to increasing the size of the air supply. For the building where these fixtures were installed, original estimates for the system utilizing the panels were slightly higher for the mechanical contract, but overall costs for the installation including structural requirements showed a considerable saving over any of the conventional systems considered. All too frequently, unit costs based on general assumptions are misleading; almost every building is subject to individual engineering analysis.

The curve illustrating the heat transfer from the top of the fixture is shown in the Appendix. It indicates that with effective temperature differences over 9½ F deg, which would be the normal operating range, the fixture absorbs heat from the ceiling space. At zero degrees effective temperature difference, approximately 75 Btu per fixture, or approximately 22 percent of the electrical input is rejected to the ceiling space. This amount is probably greater than for a conventional fixture because of the added radiating surface of the panel extensions. Further examination of the data discloses that most of this energy would be re-radiated and convected downward through the ceiling.

Mr. Schutrum's comparison of the results of this paper with the ASHAE Laboratory findings on the performance of full panelled ceilings is very gratifying. He points out

that for the conditions mentioned, the cold light system requires less air and less critical water temperatures than the other method. Attention was called to the fact that the slope of the overall performance curve shown in Fig. 9 would vary if the resistance of the panel were changed. This is quite true, but the statement should not be interpreted to mean that the slope of this curve bears any fixed relationship to panel resistance—primarily because the curve was based essentially on surrounding air temperature rather than panel surface temperature. Additionally, it would be impossible to assign a valid effective area to the panel fixture in order to arrive at a unit resistance, nor would it be possible to arrive at a valid average surface temperature. It is feasible to compute the resistance of small areas on the fixture; most of these turn out to be of much smaller magnitude than unity. It should be stressed that the data are valid only for these specifically designed fixtures, and would be meaningless if applied to luminaires of different construction or different tube bonding.

It was noted that in the paper, Table 4 and Fig. 8 did not agree. This was primarily because the chart was taken from data averaged over a greater space with a different general supply air temperature because of temperature rise. Minor corrections have been made, and the two are now compatible.

With regard to the question of greater light intensities, no types of fixtures were studied other than those described in this paper. The results are intended to demonstrate the general mechanisms involved in the heat transfer process, and give accurate data for just one type of luminaire. Heavier intensity fixtures have been used in other buildings, such as the Imperial Oil Building in Toronto, and they have been found to be as successful. It is hoped that these methods will lead to the testing of other types of luminaires.

Mr. Howarth inquired as to what credit was given to the horizontal panel extensions. These flat panels will do considerable *work*, as shown in Fig. 7. With effective temperature differences of over 10 F deg, the fixture as a whole will absorb more heat than the equivalent of the electrical input. There was no evidence of any transverse heat flow from the ceiling pans into the water cooled panel extensions of the fixture.

In answer to Mr. McKinley's question, the system installed at Manufacturers Life Insurance Company is definitely a combined system. As previously stated, if the perimeter losses of a large, multi-story building are counteracted, the interior requires cooling regardless of outside temperature. The question always arises as to how far in from the wall should heat be provided. In this building, warm water is circulated through panels under the windows and, in extreme weather, through flat panels in a 3-ft strip of ceiling adjacent to the outside wall. The original design included a series of *cold lights* on a separate circuit in a 16-ft exterior zone to be separately controlled. Continued complaints of stuffiness made it necessary to operate even these luminaires on the same temperature water supply as the interior; this supply temperature is uniformly cold whenever the building lights are on. To confirm the validity of this type of control, a thermocouple was buried in the center of a concrete floor slab and the leads were run to a permanent strip chart recorder. The temperature of the slab did not vary more than a fraction of a degree from August to February. On the other hand, there is no reason why the *cold lights* could not be used for winter heating in a truly exterior zone if the water temperatures were held sufficiently low that there would be no harmful effect to the electrical components.

1641

No. 1641

PULSATIONS IN RESIDENTIAL GAS FURNACES WITH MULTIPLE-PORT BURNERS

By A. A. PUTNAM*, COLUMBUS, OHIO

This paper is the result of research sponsored jointly by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, the *American Gas Association* and the *Oil-Heat Institute, of America, Inc.,* conducted through Battelle Memorial Institute.

THIS IS the third of a series of reports presenting results obtained from a research program directed toward the problems of combustion noise in oil- and gas-fired residential heating equipment. In an initial field work phase of this general research program, pulsations were observed in oil furnaces and in 2 basically different types of gas-fired furnaces, (1) those fired with single-port burners, and (2) those fired with multiple-port, slot, or ribbon burners. The oscillations associated with oil furnaces and single-port gas furnaces have been discussed in a preliminary way in previous papers[1, 2].

Studies of horizontal gas furnaces equipped with multiple-port, slot, or ribbon burners show that this general class of furnace was of a higher frequency range than that encountered in furnaces equipped with single-port burners. An explanation for oscillations generated in such furnaces has been formulated in the present research program. This information is based on information obtained from the field studies and some supplemental laboratory tests on horizontal furnaces, plus the available information in the literature.

The purpose of this paper is to outline the postulated mechanism whereby oscillations may or may not exist in furnaces fired with multiple-port, slot, or ribbon burners; this will serve as a basis for a subsequent paper on engineering applications of the theory. The postulated mechanism by which the driving of the oscillation occurs is first described and then defined mathematically. A presentation and discussion of supporting experimental data concludes the paper. Basic

---

* Assistant Division Chief, Battelle Memorial Institute.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

features of furnaces that influence pulsation characteristics are considered in some detail in the Appendix.

Mechanism of Driving

The mechanism for driving oscillations in a gas furnace using multiple-port burners, is arrived at by considering a certain type of organ-pipe acoustical model. Fig. 1 illustrates this model. The burner heat-exchanger assembly is represented by 2 concentric open-ended tubes of different diameters. The smaller tube represents the burner which is shown entering within the larger tube representing a heat-exchanger section.

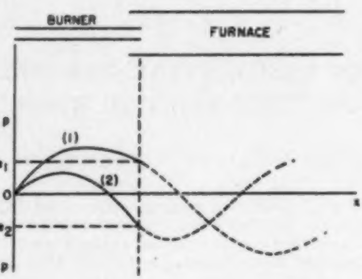


Fig. 1—Acoustical Model for Gas Furnaces Utilizing Multiple-Port Burners

If a standing wave exists in the assembly, the portion of the standing wave in the burner will, in general, take either a form similar to Curve (1) which has a zero pressure (ambient pressure) node at some point beyond the end of the burner tube, or a form similar to Curve (2) which has a node within the burner tube. As a consequence, the pressure at the end of the burner tube will fluctuate, taking on values such as p_{e1} or p_{e2} shown in Fig. 1 merely to illustrate 2 values of a time variable.

If there is a flow into the furnace at the end of the burner tube, the flow will not remain constant as there will be added a velocity fluctuation due to the fluctuation in pressure. Theoretically, the fluctuation in velocity is ¼ wavelength out of phase with the pressure fluctuation.

Fig. 2 illustrates the case where the velocity wave leads the pressure wave by ¼ cycle. The figure would not change if the lead was 5/4 cycle, 9/4 cycle, and so on. Fig. 3 illustrates the case where the velocity wave lags the pressure wave by ¼, 5/4, (and so on) cycle.

A change in the velocity of the gases means a change in the quantity of mixture available for burning and a similar change in the rate of heat release. By Rayleigh's principle, a component of fluctuating heat addition must take place in phase with the pressure wave if the oscillation is to be driven. Obviously, if burning took place exactly at the mouth of the burner and if the flame were infinitely thin, the

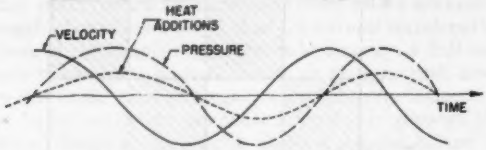


Fig. 2—Case Where The Velocity Wave Leads The Pressure Wave by ¼, 5/4, . . . , Cycles

heat addition wave would coincide with the velocity wave and not with the pressure wave and the oscillation would not be driven.

However, in an actual furnace, the flame does not attach itself flatly in the burner mouth, but at some distance away from the burner where the burning occurs over a finite distance. Because it takes time to travel this distance, there is thus introduced a *time lag* between the velocity wave and the heat addition wave which may bring the latter to coincide in time with the pressure wave as shown by the dashed curves in Figs. 2 and 3. If this occurs, oscillations will be driven; if the time-lag is sufficiently different from that needed to match the pressure wave, oscillations will not occur. Detailed discussion of the factors affecting time lag, as well as those affecting the aforementioned standing-wave shapes, is presented in the Appendix.

Even if the mentioned time-lag condition necessary to promote oscillations is met, the furnace may not oscillate if damping forces due to openings to the outside of the furnace, to bends, to friction, and like causes, exceed the force driving the oscillations. It is not surprising, therefore, that oscillations occur in only a small percentage of installations. However, even a small percentage can be extremely troublesome.

This description of events in the system will now be enlarged as a basis for developing a mathematical model of driving of oscillations. To simplify the mathematics, a sine-wave shape is assumed for the pressure and velocity waves. Past experience has indicated that this assumption, even if not true, does not affect the broad conclusions of this study. On this basis, the oscillating pressure, p, and velocity, v, along the burner from the upstream end are given by[8]

$$p = p_0 \sin (\omega x/c) \sin \omega t,$$

$$v = (p_0/\rho c) \cos (\omega x/c) \cos \omega t.$$

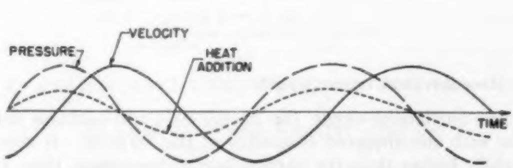


Fig. 3—Case Where The Velocity Wave Lags The Pressure Wave by ¼, 5/4, . . . , Cycles

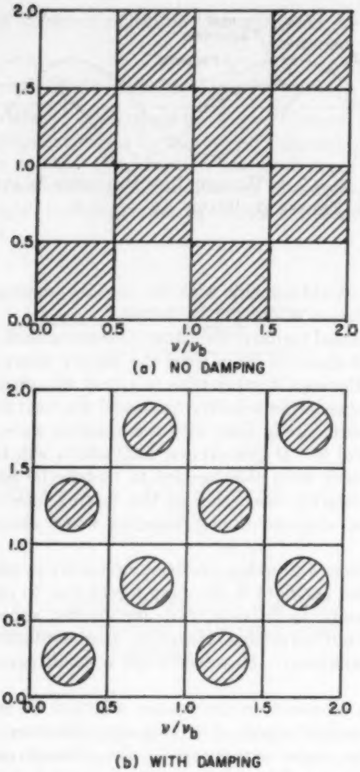


FIG. 4—REGIONS OF PULSATION IN SIMPLE MULTIPLE-PORT BURNER

At the natural angular frequency of the burner, ω_b, the pressure at the downstream end is zero. Thus

$$\omega_b L = \pi c,$$

where

L = the equivalent burner length.

When furnace oscillations occur, the burner does not oscillate with its natural frequency but with the observed frequency of the furnace. If this observed frequency is slightly higher than the natural burner frequency, then the pressure at the furnace end of the burner will vary with the pressure in the furnace, while a short distance inside the burner, the pressure amplitude will be zero. This follows

from the fact that zero pressure amplitude still exists at the inlet end of the burner, and the wavelength of oscillation is less than the natural wavelength of the burner. Thus, on the basis of the simple organ-pipe-tube analogy, a half wavelength does not reach entirely through the burner. Since air tends to rush toward the points of zero pressure amplitude from the region where pressure is decreasing, it follows that there must be an oscillating component of velocity through the burner exit. When the observed frequency is a little above a natural burner frequency, the oscillating component of the velocity at the ports leads the oscillating component of the pressure, while the opposite is true when the observed frequency is a little lower than the natural frequency. In mathematical form, the pressure at the burner exit, p_e and the exit velocity v_e, are given by

$$p_e = p_0 \sin\left(\frac{\omega}{\omega_b}\right)\pi \sin\omega t = p_0 \sin\left(\frac{\nu}{\nu_b}\right)\pi \sin\omega t,$$

$$v_e = \left(\frac{p_0}{\rho c}\right)\cos\left(\frac{\omega}{\omega_b}\right)\pi\cos\omega t = \left(\frac{p_0}{\rho c}\right)\cos\left(\frac{\nu}{\nu_b}\right)\pi\cos\omega t$$

As stated earlier, there must not be only pulses in the rate of flow of combustible gases, but these gases must burn at the right time in relation to the pressure variation in the furnace. If the oscillating component of the gas exiting from the ports leads the pressure variations in the furnace by ¼ cycle, then the best value for the time lag to produce oscillations would be about ¼ cycle. Except for the presence of complications to be mentioned later, a time lag of 1¼ cycle, 2¼ cycle, and so on, would seem equally desirable. On the other hand, if the velocity were lagging the pressure in the furnace by ¼ cycle, then the optimum values would be about ¾, 1¾, and so forth. Thus, the changeover points would occur at integer values of $2\tau/(1/\nu)$, or $2\tau\nu$, where τ is the *time lag*.

Expressing this idea more formally, driving of the oscillation will occur when an oscillating component of the heat release from the burner is in phase with the pressure in the furnace, or neglecting damping forces

$$\int_{cycle} hp\,dt > 0$$

Since the pressure in the furnace at the burner is p_e, and the rate of heat release lags the injection of the mixture from the ports by the time lag, τ, so that

$$h_t \sim v_e(t-\tau)$$

the inequality becomes

$$\int_{cycle}\left[\left(\frac{p_0}{\rho c}\right)\cos\left(\frac{\nu}{\nu_b}\right)\pi\cos\omega(t-\tau)\right]\left[p_0\sin\left(\frac{\nu}{\nu_b}\right)\pi\sin\omega t\right]dt$$
$$=\left(\frac{p_0^2\pi}{2\rho c\omega}\right)\sin\omega\tau\sin 2\pi\left(\frac{\nu}{\nu_b}\right) > 0$$

Since $(p_0^2\,\pi/2\,\rho\,c\,\omega)$ is always positive, driving may occur whenever

$$\sin\omega\tau\sin 2\pi\left(\frac{\nu}{\nu_b}\right) = \sin 2\pi\nu\tau\sin 2\pi\left(\frac{\nu}{\nu_b}\right) > 0 \text{ -- (1a)}$$

or

$$\sin 2\pi\left(\frac{\nu}{\nu_b}\right)(\nu_b\tau)\sin 2\pi\left(\frac{\nu}{\nu_b}\right) > 0 \text{ ------- (1b)}$$

Fig. 4a is a map of the oscillating regions of a simple burner, as deduced from the inequality, Equation 1a. The regions in which oscillations can occur are crosshatched. Where there is no crosshatching oscillations cannot occur. Departures from the hypothesized simple burner shape will change the positions of the vertical lines, while departures in shape of the flame from a simple flat shape will change the positions of the horizontal lines.

No consideration has as yet been given to the effect of acoustic damping forces in a furnace. These forces are ever present in wall losses, losses from sudden expansions and contractions, and end losses or acoustic radiation to the surroundings. A complete consideration of these losses is beyond the scope of this paper, and could not in any case be related to a particular furnace design with the present status of knowledge. However, it is clear that such losses will decrease the range of variables in which oscillations are possible in a furnace. Mathematically, such damping forces are provided for by replacing the zero of the inequality of Equations 1a and 1b by a positive term, the value of which depends upon (*a*) the manner in which the damping forces are produced and (*b*) the observed frequency of the furnace.

Fig. 4b shows the effect of such damping losses on the results of Fig. 4a. As the losses increase, or the driving forces decrease, these cross-hatched areas would decrease until they vanish. It might be mentioned that the fact that the flame is not flat causes the areas to decrease in size also, as $\nu\tau$ increases.

This general description of the theory has been presented to give an overall grasp of the problem of predicting the ranges of variables for which oscillation can occur in furnaces fired with multiple-port, slot, or ribbon burners; there is no intention to imply great rigor in the argument. Thus, Fig. 4 and subsequent similar figures should not be used for particular cases without a full appreciation of their limitations, and the assumptions upon which they are based.

SPECIFIC USE OF THEORY

Although Fig. 4 follows directly from the application of Equation 1a, and is readily understood, the pair of dimensionless groups involved in the coordinates, $\nu\,\tau$ and ν/ν_b, are not the most convenient for comparison with experimental data. On the other hand, an alternate dimensionless group, $\nu_b\,\tau$, combines only parameters related to the burner. From Equation 1b it can be seen that this group can be used equally well with ν/ν_b to define the oscillating and nonoscillating regions. However, for the new terms a log-log plot now makes a more convenient coordinate system.

Fig. 5 is such a figure, corresponding to Fig. 4b, and showing a development from Equation 1b of the type of log-log plot just mentioned. The ranges of variables in which driving of an oscillation is potentially possible are delineated by the diagonal lines, A, B, C, D, . . . and the vertical lines at 0.5, 1.0, 1.5, . . . and may be identified by the patterns of included boundary lines in the lower left portion of the figure, and by the crosshatched areas in the upper right-hand portion. It is seen that the ranges of variables of potential driving alternate in a checkerboard pattern with ranges for which the oscillations will not occur. For an actual burner, as contrasted to the idealized shape used in the foregoing mathematical derivation, the positions of the vertical and diagonal lines and corresponding ranges of variables would be shifted, but the alternating characteristic of the patterns would not change. With damping present, the ranges of variables for which driving will occur are smaller; the regions enclosed by the boundary lines labeled 1, 2, and 4 on Fig. 5 are an example of the effect of increasing the damping. Likewise, if the

damping effects are constant, the ranges of variables for which the driving will occur decrease in size in the order 1, 2, 4 as the input of driving energy decreases. The numerical designations of the boundary lines surrounding driving regions in Fig. 5 are such that, for constant damping, the product of these numerical designations by the driving output remains constant for a given system. For example, consider a furnace in which the ratio of driving to damping forces is such that oscillations occur when $\nu_b\tau$ and ν/ν_b are such that the operating point is within

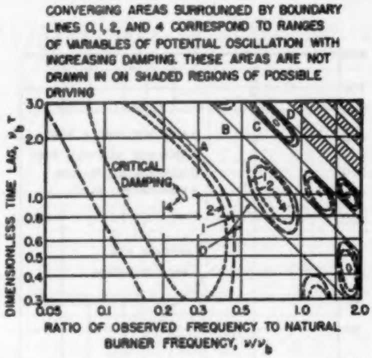


FIG. 5—RANGES OF VARIABLES FOR PULSATION PRODUCTION IN A FURNACE FIRED WITH A MULTIPLE-PORT GAS BURNER PREDICTED FROM THEORETICAL CONSIDERATIONS

boundary line 2 enclosing the point (0.25, 1.0). Now if the heat-release rate of the primary flame is doubled, the region in which oscillations are possible will expand to that enclosed by boundary line 1.

As one passes successively upward or to the right across the 45 deg diagonals designated by A, B, C . . . , the effect of the deviation of the flame from a flat shape becomes increasingly important. For constant time lag, this deviation is related to the port Reynolds number. Thus, for increasing values of port Reynolds number, of $\nu_b\tau$, or ν/ν_b, a lesser portion of the oscillating heat input is devoted to driving; this may be considered the equivalent of an increase in boundary-line number.

The amplitude of oscillation is expected to be related somewhat to the difference in the operating values of the parameters and the values at the corresponding boundary line. However, the non-linear character of the oscillating mechanism precludes any exact prediction of amplitude.

EXPERIMENTAL RESULTS

Fig. 6 shows all the gas furnace frequencies observed during the field tests conducted in the initial phase of this research. The observed frequencies for 3 of the

4 horizontal furnaces were about 600 cps. The frequency for the fourth unit was only 135 cps, although from the length of flue-gas travel, it was expected to have the highest frequency. One of the high-boy units had a multiple-port burner, but its frequency was found to be in line with those of the other high-boy units fired with single-port burners. However, the mechanism of driving in the high-boy unit equipped with the multiple-port burner is the same as for the horizontal units, as a consequence of the similarity in burners. Two of the horizontal furnaces tested, the propane-fired unit with the 620 cps frequency, and the natural gas-fired unit with the low frequency of 135 cps, were studied more extensively at a later

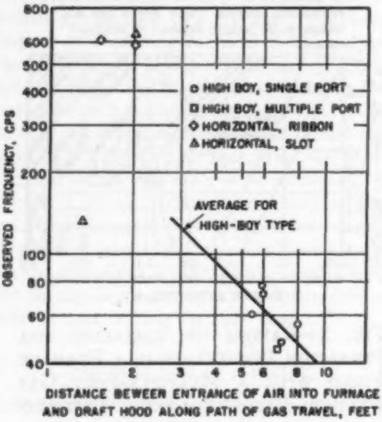


FIG. 6—FREQUENCY OF OSCILLATION AS A FUNCTION OF FURNACE SIZE

date; additional tests also were run on the burners separate from the furnaces from all 4 horizontal units and on the multiple-port high-boy burners.

Table 1 is a compilation of the pertinent data obtained in all tests. Six different burners and 5 different furnaces were studied, but extensive tests were made only on furnaces I and IV. Burner frequencies were measured when they were in open space and then corrected for an assumed air-fuel ratio composition in the burner of 65 percent of stoichiometric. The firing rate was taken from the nominal rating of the burner, and corrected for changes in manifold pressure and in size of orifice spud. The observed frequency was taken from a tape recording of the furnace if oscillation occurred. When no oscillation occurred, the furnace frequency was assumed to be the same as under oscillatory conditions. The fuel parameters used in computing the time lag were obtained from Table A-1 in the Appendix.

Propane-fired horizontal furnace I would not generate oscillations with burner A. With burner B, overfiring by increasing the orifice spud size and the manifold pressure on all burners produced an oscillation of about 620 cps in the heating-up period, immediately after a cold start; usually the oscillations did not persist after

TABLE 1—RESULTS OF TESTS ON VARIOUS BURNER-FURNACE COMBINATIONS

Burner											Furnace				
Burner Type	Normal Fuel	Meas Freq	Test Fuel	Corr[b] Freq	Firing Rate, Btu/hr	Port Loading Btu/hr, in.^{2}	$10^3 (\delta/V_p)$	10^3 (a/nF)	$10^3 \tau$	$\nu_b \tau$	Furnace No.	Obs Freq	Freq Ratio	Oscillation	Symbol on Fig. 7
A-Ribbon	Propane	650	Propane	635	22,000	8,400	1.88	0.70	2.58	1.64	I	620[c]	0.976	No	○
					27,000	10,300	1.53	0.70	2.23	1.42	I	620[c]	0.976	No	○
					27,000	10,300	1.53	0.70	2.23	1.42	II	580[c]	0.913	No	○
					33,000	12,500	1.26	0.70	1.96	1.24	II	580[c]	0.913	No	○
B-Ribbon	Propane	560	Propane	546	22,000	8,400	1.88	0.70	2.58	1.41	I	620[c]	1.136	No	○
					27,000	10,300	1.53	0.70	2.23	1.22	I	620	1.136	Yes	●
					27,000	10,300	1.53	0.70	2.23	1.22	II	580[c]	1.062	No	○
					33,000	12,500	1.26	0.70	1.96	1.07	II	580	1.062	Yes	●
C-Milled Slot	Nat gas	637	Mfg gas	1010	45,000	19,500	0.405	0.90	1.3	1.31	III	650	0.694	Yes	◆
			Nat gas	654	45,000	19,500	0.585	2.92	3.5	2.29	III	600[d]	0.923	No	[g]
D_1-Milled Slot	Mfg gas	694[a]	Nat gas	713	42,000	31,000	0.364	1.61	2.07	1.48	IV	135	0.189	No	□
					68,000	50,000	0.223	1.61	1.83	1.30	IV	135	0.189	Yes	■
					42,000	31,000	0.364	1.61	2.07	1.48	IV	650[e]	0.91	No	□
					68,000	50,000	0.223	1.61	1.83	1.30	IV	650[e]	0.91	No[f]	□
D_2-Milled Slot	Nat gas	694	Nat gas	713	42,000	19,000	0.602	2.92	3.52	2.50	IV	135	0.189	No	□
					68,000	30,200	0.371	2.92	3.29	2.35	IV	135	0.189	No	□
					42,000	19,000	0.602	2.92	3.52	2.50	IV	650[e]	0.91	No	□
					68,000	30,600	0.371	2.92	3.29	2.35	IV	650[e]	0.91	No	□
E-Drilled Port	Nat gas	520 505 528	Nat gas	532 avg.	105,000	26,000	0.436	4.03	4.47	2.38	V	45	0.084	Yes	▲

[a] Value assumed same as for natural gas burner.
[b] Assuming 65 percent of stoichiometric.
[c] Value obtained from corresponding furnace where oscillation occurred.
[d] Extremely weak and transient note.
[e] Value obtained from Furnace III with same length of heat exchanger tubes.
[f] It should be considered that this oscillation might have occurred at 135 cycles.
[g] Indicates open diamond on Fig. 7.

the furnace became warm. The observed oscillations were not regular, in that both the frequency and the amplitude changed during the heating-up period.

Propane-fired horizontal furnace II, a larger version of furnace I, used the same type of burner as furnace I but at a higher rating. The observed frequency, using burner B and when overfiring, was somewhat less than that of furnace I. No oscillation was obtained in furnace II with burner A even when overfiring.

Furnace III was a moderate-sized horizontal furnace equipped with milled slot natural-gas burners. By using manufactured gas, this furnace could be made to oscillate intensely. Using natural gas, however, only a weak and transient oscillation that was barely detectable could be produced.

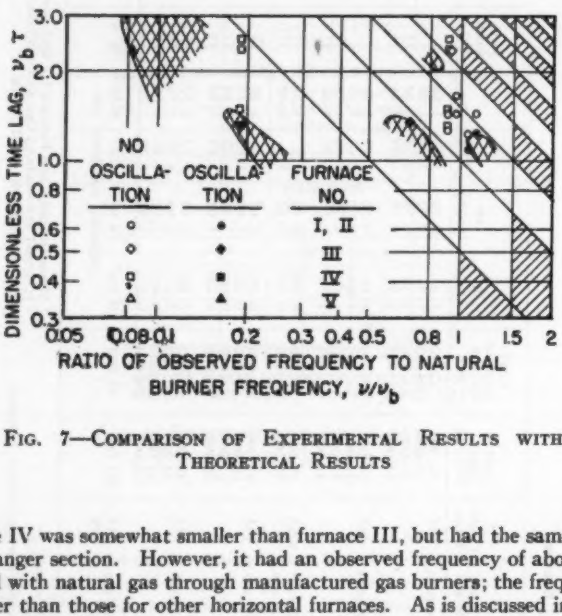


FIG. 7—COMPARISON OF EXPERIMENTAL RESULTS WITH THEORETICAL RESULTS

Furnace IV was somewhat smaller than furnace III, but had the same length of heat exchanger section. However, it had an observed frequency of about 135 cps when fired with natural gas through manufactured gas burners; the frequency was much lower than those for other horizontal furnaces. As is discussed in more detail in the Appendix, this resulted from the furnace acting as a double-necked Helmholtz resonator.

Furnace V was a home installation of an older design of high-boy equipped with 3 drilled-port burners which, in combination, formed a circular grid.

CORRELATION OF EXPERIMENTAL RESULTS WITH THEORY

In the discussion of the relation of the experimental results to the general theory that has been derived, sets of data will be considered as they pertain to the clarification of certain specific points. Thus, the order of presentation used in Table 1 will not necessarily be followed herein.

Fig. 7 is a replot of pertinent portions of Fig. 5, with selected data points from Table 1 superimposed. Positions of the boundary lines indicating the apparent

relation of driving to damping forces for each set of data are sketched in, rather than the sets of boundary lines in Fig. 5. In regions where theory indicated no oscillations should occur, none was found. In regions where theory indicated oscillations were possible, several instances were observed in which they occurred. These later data will be discussed next.

Fig. 8 affords a useful basis for considering each pair of data points between which the firing rate was changed, but the furnace frequency and burner frequency were constant. Such a firing rate change results in moving the furnace operating point along a vertical line in Fig. 5 or 7, as $\nu_b\tau$ is changed and ν/ν_b is held constant. If this change in $\nu_b\tau$ is brought about by a change in the flow rate while maintaining the same mixture ratio, then, as shown in Equation A-1 in the Appendix, only the

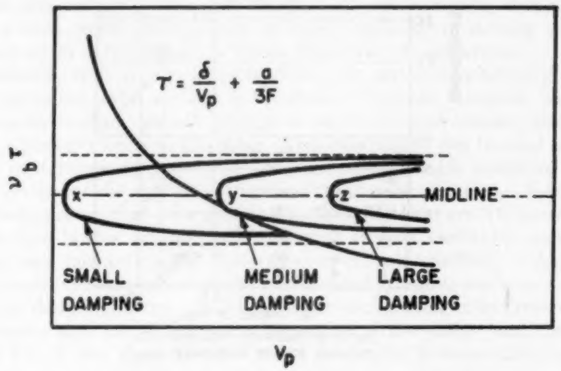


FIG. 8—EFFECT OF CHANGE IN FIRING RATE ON PRESENCE OF OSCILLATIONS WITH ν/ν_b FIXED

factor V_p in τ varies while ν_b, and the other factors in τ, remain constant. On the other hand, as the mixture flow rate increases, the energy release for driving the oscillation increases and thus the range of variables within which oscillations occur increases. Thus, one may sketch a series of curves such as shown in Fig. 8 for 3 degrees of acoustic damping. Considering as an example a value of $\nu/\nu_b \cong 0.6$ on Fig. 5, the dashed lines of Fig. 8 would correspond to lines A and B of Fig. 5 where they intersect $\nu/\nu_b = 0.6$, and Curves x, y, and z of Fig. 8 would correspond in a sense to boundary lines 1, 2, and 4 of Fig. 5.

Fig. 9 presents several sub-plots of Fig. 7, for purposes of comparison with the curves of Fig. 8. For instance, Fig. 9a presents the data which correspond to the case associated with Curve x of Fig. 8. For Curve x, wherein the relative damping is low, oscillation will occur over a wide range of $\nu_b\tau$, practically the entire possible range of oscillation; this case is illustrated in the sense of increasing V_p, by the change on Fig. 9a along $\nu/\nu_b = 1.136$ from $\nu_b\tau = 1.41$ to 1.22. Another example of the Curve x condition is shown by the series of points along $\nu/\nu_b = 0.189$, leading to a Helmholtz-type oscillation in furnace IV.

In the case of Curve y of Fig. 8, the relative damping is larger, and only a small region of oscillation is indicated. This region, it may be noted, is entirely below

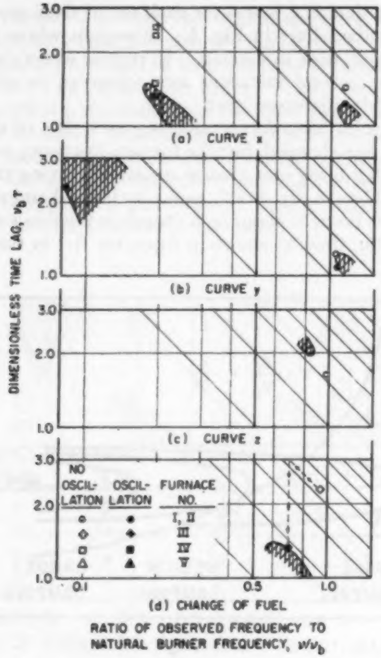


FIG. 9—SUB PLOTS OF FIGURE 7

the midline in the potential oscillating range. It is believed that such a condition is involved in the case of the passing along the line on Fig. 9b represented by $\nu/\nu_b = 1.062$ from $\nu_b\tau = 1.22$ to the oscillation point. The datum point on Fig. 9b from furnace V, at $\nu/\nu_b = 0.0184$, is also believed to correspond to case y.

In the case of Curve z of Fig. 8, the damping forces are large enough that no oscillation occurs. This probably is illustrated by the datum point for furnace I on Fig. 9c.

The data points of Fig. 9d, obtained from furnace III, conform to the expected patterns of results, but are also interesting because they show the effect of change of fuel. The first effect of change of fuel is to change the natural frequency of oscillation of the burner. If the fuel is changed to one of less density, such as changing from methane to manufactured gas, the frequency of the burner increases. With no change in time lag, this would move the operating point along a 45 deg slope on Fig. 9d, upward to the left. For the same heat-release rate, the volume flow rate does not change much. However, when the burning velocity of the fuel increases, as in changing from methane to manufactured gas, the dark space can be expected to decrease. Both of these factors cause a decrease in the value of

τ. Thus, the operating point moves downward. The net result in changing from methane to manufactured gas is shown by the positions of the two diamonds on Fig. 9d.

CONCLUDING REMARKS

A logical basic mechanism of combustion-driven oscillations in furnaces equipped with multiple-port, slot, or ribbon burners has been described. This mechanism is based on a considerable amount of observation reported in the literature. Test results obtained in this program are also in agreement with the theory, and this tends to support this application of the theory. However, several factors which are incompletely understood or upon which there are insufficient data affect the occurrence of oscillations. One factor involved is the time lag, and the manner in which it is affected by burner-port design, fuel, mixture ratio, and mixture flow rate. A second factor concerns the relative magnitude of driving and damping terms involved in determining the stable amplitude of oscillations.

By laboratory tests it is possible to define the region in which a given furnace design is operating, even without a knowledge of all the foregoing factors. This can be done by making discrete changes in the burner and furnace, and estimating unknown values as closely as possible. This information can be used as a basis for judgment in determining how close a design is to unstable operating conditions, and what design factors may be changed to improve the situation. Some suggested areas for design changes and the principal factors they alter are: (1) porting design—τ, (2) effective burner length—ν_b, (3) length of heat exchanger section—ν, (4) baffling in heat exchangers and flue-ways—ν, (5) inlet baffling—ν for Helmholtz frequency, and (6) relative position of burner head and heat exchanger inlet—ratio of driving to damping forces. Which of these changes might be preferred depends on the relative position of the operating region of the design and the oscillating regions of Fig. 5, and the susceptability of the design to economical alteration of various features.

It is desired to acknowledge the assistance of W. R. Dennis, Gail M. Clough, and Carl F. Speich in obtaining various portions of the data used herein. This research is sponsored by ASHAE, the *American Gas Association* and the *Oil-Heat Institute of America, Inc.*; support within the latter organization came from the equipment manufacturers and from oil refiners. The project is guided by the Pulsation Research Steering Subcommittee of the Technical Advisory Committee on Combustion of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS. The able assistance of the members of that Subcommittee not only in guiding the work but in providing necessary information and materials is gratefully acknowledged. Special thanks are due to the personnel of the Armstrong Furnace Co. and the Surface Combustion Corp., both of Columbus, Ohio, who provided the experimental furnaces used.

REFERENCES

1. RESEARCH REPORT No. 1596—Pulsations in Residential Heating Equipment—Preliminary Results, by A. A. Putnam and W. R. Dennis (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 153).

2. RESEARCH REPORT No. 1612—Acoustic Coupling of Residential Furnaces with Their Surroundings, by C. F. Speich, W. R. Dennis and A. A. Putnam (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 413).

3. *Vibration and Sound*, by Philip Morse (McGraw-Hill Book Co., Inc., New York, 1948.

4. A Study of Burner Oscillations of the Organ-Pipe Type, by A. A. Putnam and W. R. Dennis (*Transactions of American Society of Mechanical Engineers*, Vol. 75, No. 1, 1953, p. 15).

5. Fundamental Flashback Blowoff, and Yellow-Tip Limits of Fuel Gas-Air Mixtures, by J. Grumer, M. E. Harris and V. R. Rowe (Bureau of Mines *Report of Investigation 5225*, July 1956).

6. Effect of Initial Mixture Temperature on Flame Speeds and Blow-Off Limits of Propane-Air Flames, by G. L. Dugger (NACA TN 2170, August 1950).

7. Effect of Initial Mixture Temperature on Flame Speed of Methane-Air, Propane-Air, and Ethylene-Air Mixtures (NACA TN 2374, May 1951).

8. *Explosion and Combustion Processes in Gases*, by W. Jost (McGraw-Hill Book Co., Inc., New York, 1946, p. 123).

9. Research in Fundamentals of Atmospheric Gas Burner Design (American Gas Association Testing Laboratories Bulletin No. 10, March 1940).

NOMENCLATURE

a = effective radius of port or half width of slot.
c = velocity of sound.
F = burning velocity.
h = oscillating component of the rate of heat release; flame height.
L = equivalent burner length, $c/2\nu_b$.
p = oscillating component of the pressure.
p_0 = index pressure.
p_b = oscillating pressure in furnace at burner port level.
R_e = Reynolds number.
t = time.
v = oscillating component of the velocity.
v_b = oscillating component of the velocity at burner port.
V_p = average velocity through ports.
x = coordinate.
α = constant to be evaluated from tests.
δ = dark space (distance between the top face of the burner and the bottom of the flame).
ν = frequency.
ν_b = natural frequency of burner.
ρ = density.
τ = time lag.
ω = angular frequency, $2\pi\nu$.
ω_b = angular natural frequency of burner, $2\pi\nu_b$.

APPENDIX

DISCUSSION OF IMPORTANT PARAMETERS

Three parameters, (1) the *time lag*, (2) the natural frequency of the burner, and (3) the observed frequency of the burner-furnace assembly have been shown to be of fundamental importance to an understanding of the postulated mechanism by which a multiple-port, slot, or ribbon burner can drive an oscillation in a furnace. The evaluation of these 3 parameters will now be considered in some detail in connection with a residential gas furnace. It will be found that, although the concept behind each parameter is simple, theoretical calculation of their value presents difficulties. This results from the large number of furnace or burner variables which can influence each of the 3 parameters. However, the experimental determination of the values of 2 of the parameters is possible, and was done in this work. Such an experimental determination also appears feasible for the third parameter, the time lag, which was, however, computed in this program.

Time Lag: Fig. A-1 is shown to explain the meaning of *time lag*, which is the time interval between the moment when a disturbance in flow rate at the burner face occurs and the moment the resulting disturbance in heat-release rate of the primary flame reaches a maximum. Time lag is composed of 2 parts. The first part is the time required for the flow of premixed air and fuel gas, moving at velocity V_p, to pass from the face of the burner through the dark space of thickness, δ, under the base of the flame. The second part is the time for the mixture to pass from the base of the flame to the average height of burning in the primary flame front. For a cone-shaped primary flame, this average height can be shown[4] to be about ⅓ the height, h. Since the ratio of h to a is approximately that of V_p to F, the height, h, is approximately equal to

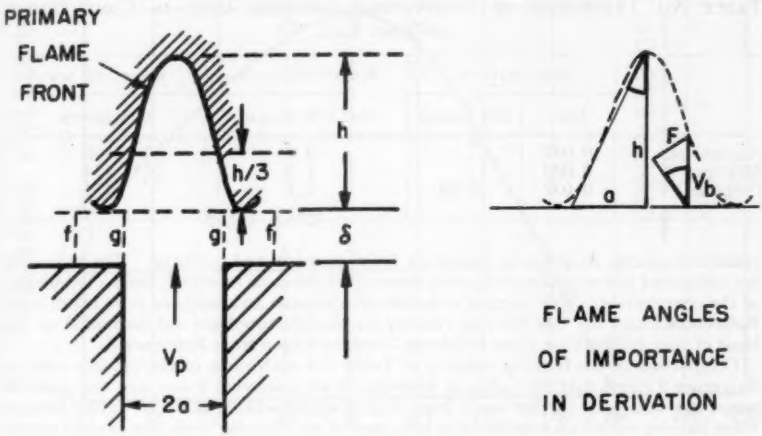


FIG. A-1—SKETCH OF CONICAL FLAME

$a(V_p/F)$ where a is the radius of the port and F is the burning velocity. The time lag, τ, for a circular port is thus

$$\tau = \frac{\delta}{V_p} + \frac{a}{3}\left(\frac{V_p}{F}\right)\left(\frac{1}{V_p}\right)$$

or

$$\tau = \delta/V_p + a/3F \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{(A-1)}$$

For wedge-shaped flames over slots, it can be shown that the factor 3 would be replaced by 2, if a is taken as the half-width of the slot.

In theory, for a given burner, the time lag can be computed as a function of fuel flow rate, type of fuel, and mixture ratio. However, when the burner is operating in an ambient atmosphere of air (as in a furnace), the mixture in the dark space will vary from lean on the ambient air side, at f in Fig. A-1, to rich toward the port, at g. Thus, the thickness of the dark space used in the computation of time lag should probably be the minimum that can be obtained in the literature for the particular fuel used. Occasionally with drilled port and milled-slot burners and always with ribbon burners,

the individual flames are close enough to have interacting effects. This will tend to decrease, δ, and increase F, but the magnitude of the effect is unknown. Thirdly, the burner around the base of the flame becomes hot. This will cause a decrease in δ. The increase in air viscosity caused by heat transfer from the burner can decrease the laminar flow of primary air, and thus decrease V_p, F, and rate of heat release related to the acoustic driving. In spite of these complications (some of which cancel one another), satisfactory theoretical computations of time lag can sometimes be made, as indicated by the previous discussion in the text. Nevertheless, it is apparent that an experimental determination of this factor would be generally preferable to a theoretical computation.

Table A-1 lists the values which were used in the computation of the time lag parameter presented in Table 1. Dark space was taken as half the quenching distance between flat plates, as given in Reference 5 for propane and methane. The value for manufactured gas was estimated, using material in the same reference, from a knowledge of the composition. The burning velocities for propane and methane were taken from References 6 and 7. The burning velocity for manufactured gas was estimated on the basis of that for methane from Reference 7 and hydrogen from Reference 8.

TABLE A-1. PROPERTIES OF COMBUSTIBLE MIXTURE USED IN COMPUTATION OF TIME LAG

Fuel	Dark Space, in.		Burning Velocity,	Btu/cu ft, min.
	Min.	65% Stoich.	fps, 65% Stoich.	65% Stoich.
Natural gas	0.039		0.4	140
Mfg. gas	0.029		1.3	134
Propane	0.032	0.05	1.1	151

Comparison of the burning velocity of Table A-1 with those of the pioneer work of Reference 9 shows that the values of Reference 9 are somewhat lower, and the value for propane is relatively farther away from that of manufactured gas. However, because these burning velocities contribute to only part of the time-lag term, the general results reported herein would not be affected by using the values from Reference 9 rather than those given in Table A-1.

As an example of the procedure used in calculating the time lag for a burner, the burner used in the high-boy furnace that was tested will be considered. The entire burner consisted of 3 separate sections forming a circular grid when placed together. In the combination, there are 380 No. 32 ports, which are 0.116 inch in diameter. The firing rate is 105,000 Btu per hr. From Table A-1, the values of dark space, burning velocity, the heat-release rates per unit volume, for a 65 percent stoichiometric mixture of natural gas and air, are 0.039 in., 0.4 fps, and 140 Btu per cu ft, respectively.

Using those values, the average velocity through each port is given by

$$V_p = \left(\frac{105{,}000}{3600}\right) \Big/ \left(\frac{140}{1728}\right) (0.116)^2 \left(\frac{\pi}{4}\right) (380) = 89.5 \text{ ips}$$

Thus,

$$\delta/V_p = 0.039/89.5 = 0.436 \times 10^{-3} \text{ sec}$$

The second term in the time lag is given by

$$a/3F = 0.116/(2)(3)(4.8) = 4.03 \times 10^{-3} \text{ sec}$$

The time lag is, therefore,

$$\tau = (0.44 + 4.03)\, 10^{-3} = 4.47 \times 10^{-3} \text{ sec}$$

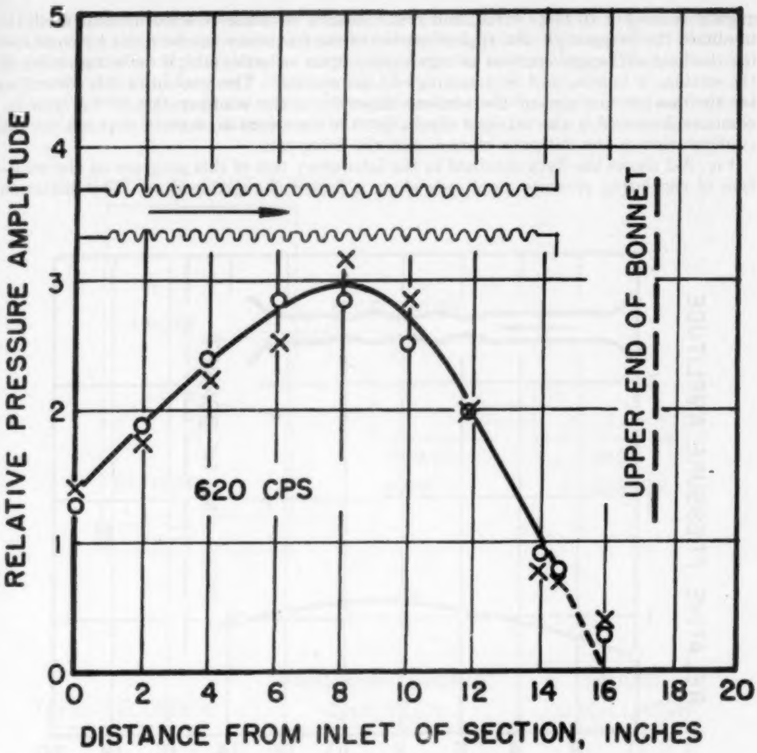


FIG. A-2—TWO SETS OF SOUND PRESSURE DATA TAKEN IN HEAT EXCHANGER OF HORIZONTAL FURNACE I WITH BURNER B

Most analyses of combustion-driven oscillations are based upon the assumption of a time lag, τ, defined in a manner similar to that just outlined. Such a procedure, which really assumes a flat or button-like flame, is acceptable in many instances, and was used in the preceding theoretical derivation. However, it should be noted that as the flame becomes less flat, this assumption becomes less tenable. The effects of the simplifying assumption just indicated have been considered qualitatively in previous portions of this paper.

The success in this study of the computational method in determining time lag is believed to result at least partially from a fortuitous choice of furnaces. The damping forces were sufficient that departures for a button-shaped flame were not important in the instances when they occurred. Other possible difficulties were also at a minimum because of the particular configurations.

FREQUENCY OF THE BURNER-FURNACE ASSEMBLY

The natural frequency of the furnace and burner assembly is an important factor in any study of pulsating furnaces. However, the theoretical computation of this fre-

quency is subject to large error, and resort should be made to experimental methods to obtain the frequency. An approximation of the frequency can be made by considering the heat exchanger sections as organ pipes open at both ends[1], if the temperature in the sections is known, and by assuming end corrections. The amount of this correction for the downstream end of the sections depends on the configuration of the plenum chamber into which the sections discharge; the upstream correction depends on the configuration of the entrance plenum and the firing rate.

Fig. A-2 shows the data obtained in the laboratory test of this program on the variation of oscillating pressure amplitude along a heat-exchanger section. The distortion

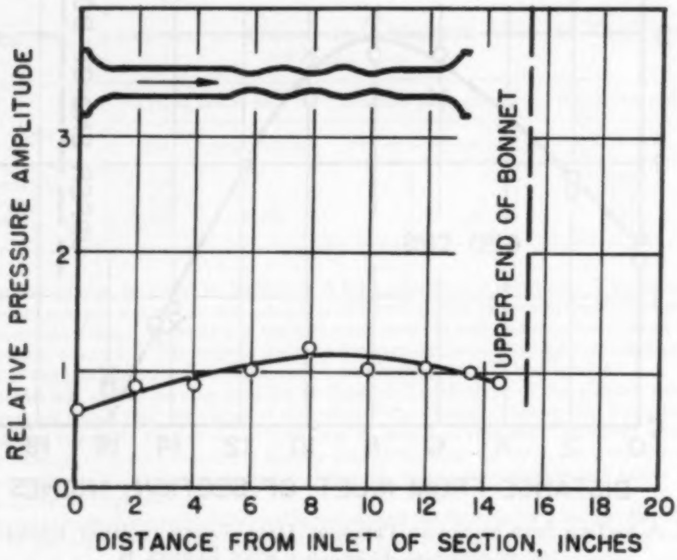


FIG. A-3—SOUND PRESSURE PATTERNS IN HEAT EXCHANGER OF FURNACE IV WITH BURNER D_1

in the amplitude pattern from sine wave form is caused by the decrease of temperature in the direction of flow. The downstream end of the section was found to act acoustically as an open end, with a small end correction of about 1.5 in., as indicated by the dotted line, even though the section discharged into a confined space. The end correction at the upstream end, where the burner fired into the heat-exchanger section, was larger than that of the downstream end. Other tests indicated that this upstream end correction increased somewhat with the rate of firing.

This test program has shown that a second, lower frequency can also be generated in some horizontal types of furnaces. In this mode of oscillation, the plenum and heat-exchanger elements act as a volume with a neck at each end, in the manner of the double Helmholtz resonator[1]. The frequency is considerably lower than the frequency of the heat-exchanger sections alone. A computation of the natural frequency in this instance would have been even more difficult to make than for the higher frequency.

Fig. A-3 shows the sound-pressure level obtained experimentally in a heat-exchanger section when a furnace is behaving as a double-necked Helmholtz resonator. It is seen that the rate of change of pressure amplitude along the heat-exchanger section is smal compared to that of furnace I shown in Fig. A-2.

Fig. A-4 indicates, schematically, the experimental technique which was used to determine the oscillating frequencies of the multiple-port gas furnace. When the furnace

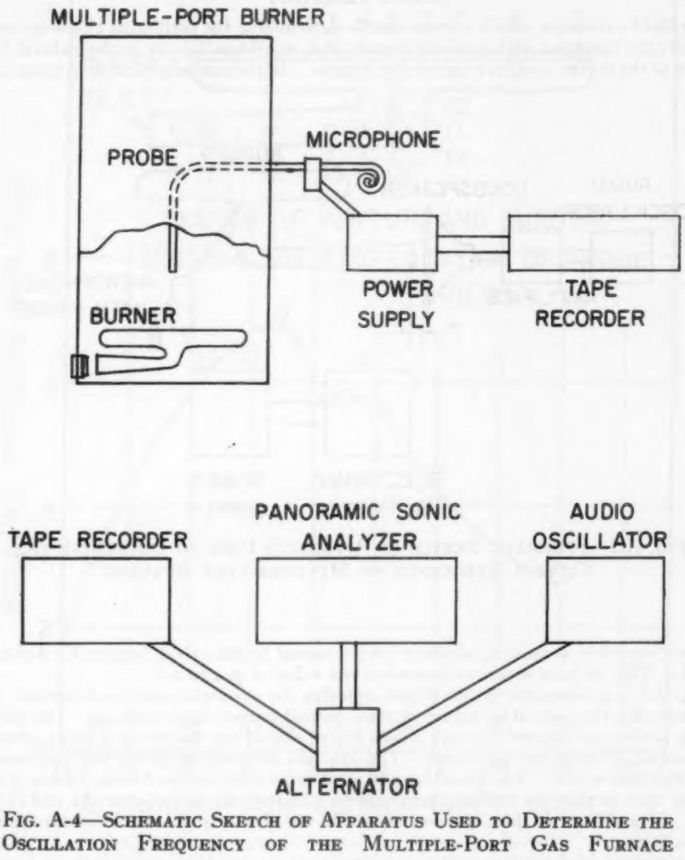


FIG. A-4—SCHEMATIC SKETCH OF APPARATUS USED TO DETERMINE THE OSCILLATION FREQUENCY OF THE MULTIPLE-PORT GAS FURNACE

was oscillating, a tape recording of the oscillation was made using a probe microphone and tape recorder. This tape was then played back and the signal from the tape fed into the alternator and then into a sonic analyzer. At the same time and in the same way, a signal from an audio oscillator was also fed into the analyzer. The purpose of the alternator was to allow the signal from the tape recorder to be displayed on the screen of the cathode-ray tube for one sweep of the analyzer and then switch the in-

puts so that the signal from the oscillator was displayed for the next sweep. The persistance of the screen of the cathode-ray tube is such that both images were displayed at one time. The frequency of the signal from the audio oscillator was varied until the two images coincided at the highest peak of the signal from the tape recorder. The oscillating furnace frequency was then equal to the known frequency of the signal from the audio oscillator.

BURNER FREQUENCY

The third parameter which is important in determining the oscillating characteristics of a furnace equipped with a multiple-port, slot, or ribbon burner is the natural frequency of the burner itself in an unconfined space. In the mathematical derivation, one

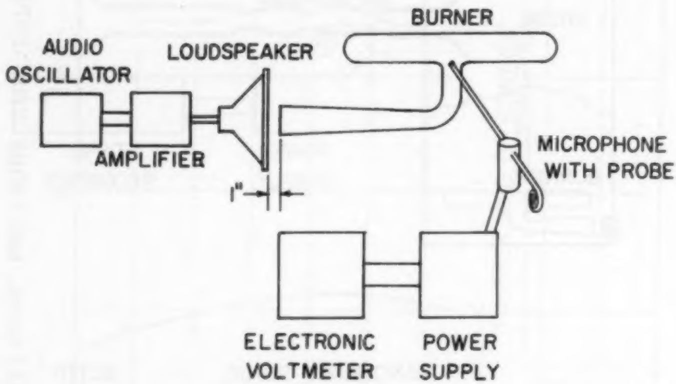


FIG. A-5—SCHEMATIC SKETCH OF APPARATUS USED TO DETERMINE THE NATURAL FREQUENCY OF MULTIPLE-PORT BURNERS

burner frequency is used to calculate the acoustical length of the burner, L, shown in Fig. 1. This, in turn serves to determine the value of p_e and v_e.

Fig. A-5 is a schematic sketch which indicates the apparatus used to determine, experimentally, the natural frequency of these unfired burners filled with air. The microphone probe was inserted through a hole in the side of the burner at a point where a high sound pressure was prevalent. The junction between the probe wall and burner wall was then sealed. The gas orifice and air damper were removed from the end of the supply tube so that the loudspeaker could be conveniently placed near the end of the tube. The absence of these components would not greatly affect the natural frequency of the burner.

Determination of the burner resonant frequency was made by varying the frequency of the signal issuing from the loudspeaker until peak value was obtained on the voltmeter. The frequency at which this maximum microphone signal voltage occurred was considered to be the natural frequency of the burner after it had been determined that none of the electrical components had natural frequencies in that range. From a knowledge of the velocity of sound in the combustible mixture relative to the sound velocity in air, this natural frequency was then corrected to give the natural frequency of the burner when containing a combustible mixture.

Fig. A-6 shows sound-pressure amplitude data for a ribbon burner installed in a furnace. This observed pressure pattern is believed representative of the pressure pattern associated with the natural burner frequency because the observed furnace

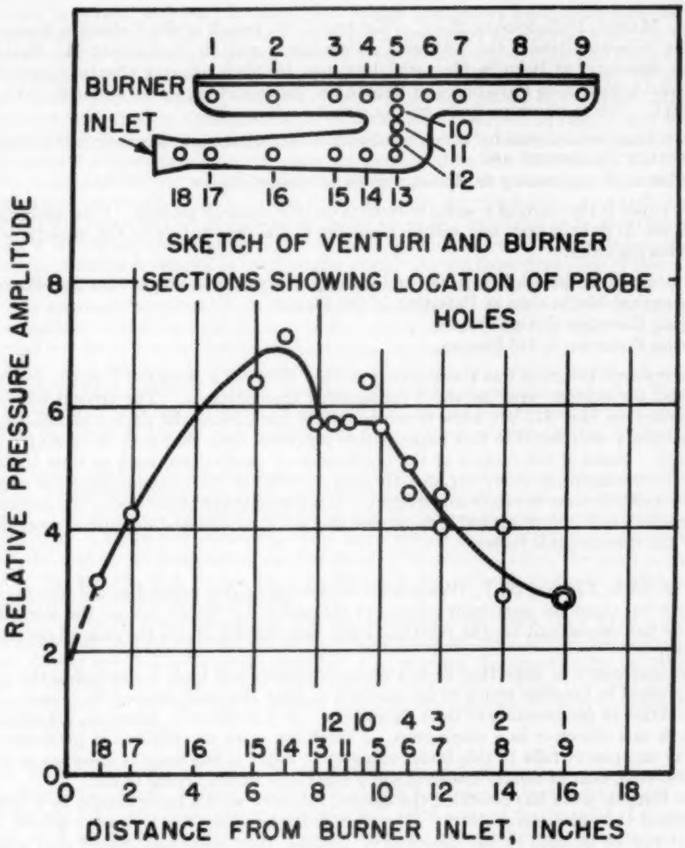


FIG. A-6—SOUND PRESSURE PATTERN IN BURNER B WITH NATURAL FREQUENCY OF 546 CPS INSTALLED IN FURNACE I OSCILLATING AT 620 CPS

frequency of 620 cps is close to the natural frequency of this burner of 546 cps. The left-hand side of the plot has a characteristic sine curve of a standing wave in an organ-pipe, with some modification due to change in cross-sectional area. There is a central

portion in the vertical section where the pressure is uniform. In the portion with burner ports, the data can be fitted by a curve of a hyperbolic cosine shape.

DISCUSSION

A. A. Marks, Philadelphia, Penn., (Written): On behalf of the Pulsation Research Steering Subcommittee of the TAC on Combustion, I wish to compliment Mr. Putnam and his associates at Battelle Memorial Institute, for their untiring efforts in pursuing the research project on Pulsation and Resonance, sponsored jointly by ASHAE, *A.G.A.*, and *OHI*. The research program was initiated to determine:

1. the basic mechanism by which combustion oscillations are driven in oil- and gas-fired heating equipment; and
2. Means of suppressing or eliminating these oscillations.

This paper is the third of a series of reports on this research project. Four additional papers are in preparation, and will be available in the near future to the Society, with the following titles.

Pulsation in Residential Heating Equipment Equipped with Single Port Gas-Burners
Theoretical Mechanism of Pulsation in Oil Flames
Curing Pulsation in Gas Flames
Curing Pulsation in Oil Flames

The research program was inaugurated in May 1954 for a period of 3 years, and was extended for another year by the 3 cooperating organizations. The project officially terminated on May 31, but work is continuing in completing the papers mentioned.

The theory postulated in this paper covers pertinent data that may be of major importance. Some of the factors of the mechanism of oscillations, such as time lag and the relative magnitude of driving and damping involved in determining the stable amplitude of oscillation are not fully understood. It is hoped that at least one of the cooperating sponsors will find it advisable to pursue this work further and determine more definitely the questionable factors.

R. W. Sage, Linden, N. J., (Written): In discussing this paper I would like to concentrate not upon the particular subject of the paper but rather discuss the work that Battelle has carried out for the ASHAE, *OHI*, and the *A.G.A.* on the general pulsation problem.

It is customary to hope that such a research project will have a conclusion that can be expressed in tangible terms to its sponsors so that they can proceed to improve the construction or performance of their equipment. It is frequently, however, not possible to reach this objective in a single step. I think the work on eliminating pulsations in heating equipment falls in this latter category. What it has done is allow us to take the first step, that of clearly understanding what it is we are trying to do.

The Battelle work has provided the heating industry with a basic insight to 3 fundamentals of the pulsation problem. They are, *first* what pulsation is, how it occurs, and how it can be studied in the laboratory. *Second*, they have established that equipment variables can and do influence pulsation and some progress has been made in isolating the effects of some of these variables. *Third*, they have shown the importance of environment on pulsation. In my opinion all 3 are important basic contributions to the study of pulsation. I think, though, that it is only honest to say that these results cannot be applied directly to solve all pulsation problems in a general way, nor can they be applied easily to any specific pulsation problem.

Having in hand, however, the published data from this study, a manufacturer can start far down stream from the sources of his problem and in an easier manner proceed toward its solution. I think it can be said that a specific piece of equipment can be

designed now, not necessarily free of pulsation, but with pulsation less liable to occur than if these results were not available. Also, if pulsation does occur, it can be corrected more easily.

Perhaps this can be illustrated more clearly by reference to work we have in progress to try to eliminate pulsations in oil-burning furnaces and boilers.

Prior to undertaking any work in our own laboratories we carefully reviewed the information that Battelle had published. We also sent our technical people to Columbus to talk in person with Mr. Putnam, Mr. Speich and others associated with the pulsation project. From this preliminary work we developed an approach to the problem guided by the theories on mechanism that Battelle had worked out and using many of the experimental techniques developed by Battelle. We soon proved to ourselves that at this point it was impossible to just redesign equipment to eliminate pulsations. We were able, though, to isolate those variables most likely to be critical and we could then make various modifications to the equipment within more limited areas. In the particular furnace-burner combination we employed in the laboratory, we found at least 3 techniques that would definitely prevent pulsation. However, 2 of these were completely impractical to use in the field and while the third may ultimately be practical, it still needs further development before we would be willing to believe that we had made a significant contribution. All of the 3 techniques evolved here were consistent with the theories developed in the Battelle work. At the same time none of them should be considered as easy and direct evolutions from Battelle work.

This approach to pulsation studies can be used by others involved in the design and development of heating equipment. It eliminates the need for this basic research being carried out by each party interested in developing corrective measures. Because of the expense, very few corporations would be in a position to undertake such an extensive program of basic research. Here, however, the results are available to all. Therefore, I feel we must consider the program as successful and as having met at least part of its initial objective of providing a solution to the problem of pulsation in heating equipment.

I feel the need for additional work at Battelle is not pressing. The real need now is for equipment manufacturers to give attention to their own specific equipment combinations. This attention should be guided by the Battelle work and the theories developed therefrom. I think I can safely say it is not a simple job to eliminate pulsations even with the additional information now at hand. However, I think the time is right to tackle the specific jobs rather than to continue the general approach which has, of necessity, been employed at Battelle.

I would like to congratulate the Battelle people on the work they have conducted and would like to take this opportunity to thank them for their help in the work that we have carried out in our own laboratories. I appreciate the opportunity to have been able to discuss their work before this audience today and I hope that in the not too distant future we will be able to appear again on the program to offer additional help in this important area.

W. B. KIRK, Cleveland, Ohio: I certainly concur with the written comments of Mr. Marks and Mr. Sage. I think the Battelle people have done a very neat job of framing the factors that do contribute to this problem. By way of testimonial, I happen to know of 2 gas appliance manufacturers who have benefited from information made available through this work.

I agree with Mr. Sage, that the work has gone as far as it should go at Battelle, and its application is then up to the manufacturers of equipment. It seems very obvious that little can be done about the time lag as described by Mr. Putnam. In the first place, even though he attempted to consider the burning of gas flames as uniform across the various ports, it is almost impossible of attainment in today's practice. For that reason, I think Battelle is to be congratulated on simplifying the problem down to the point where certain measurements can be made rather simply to define the areas in which corrections are needed. I know that they have outlined a procedure of recom-

mended steps to try. As I stated before, I know of 2 instances where this has been successful.

AUTHOR'S CLOSURE: I want to thank Messrs. Marks, Sage, and Kirk for their kind comments. Mr. Marks gave a short review of the program which he has ably guided as Chairman of the Pulsations Steering Committee, and indicates the general areas to be discussed in the 4 remaining papers covering this work.

Mr. Sage's remarks on his company's program of study of combustion noise, and its relation to this work, are greatly appreciated. This is a typical example of how a company can take advantage of the background and experience of the Battelle staff concerning factors which are beyond the scope of the usual report. We feel, as do the commentators, that the point has been reached where individual companies must be concerned with their own designs. We are only too happy to pass on our experience to them, as we have done for Mr. Sage's company, to aid in pushing their programs forward.

Before leaving the discussion of oil-fired heating units, we will use such units as an example to show why each company must consider its own specific design, and not expect a fundamental program to supply specific *cures*. Most oil-fired units have at least 2 possible modes (or frequencies) of oscillation, and 2 possible manners in which the flame can drive the oscillation. This gives 4 combinations. Considering a high-amplitude oscillation as somewhat like a flat-topped hill on a plot of the various parameters, we find there are 4 hills. Requiring a specific set of *cures* that work for every condition is like requiring that at any place in this range of hills one can get to a valley in the best possible way by heading in only one direction, for example, south. This is obviously not true. The moral is that each manufacturer must know where his unit is operating in this multitude of possibilities, before he can pick the best *cure* for his pulsation problem.

Returning to the gas-fired units, I should like to comment further on the time lag. In the paper, a simple concept of time lag was used. In an appendix to the final paper on gas-fired units, we shall present an elaboration of this concept. However, a consideration of available information indicates that factors have been neglected which may be just as important as those considered, in this further elaboration. To attempt to include all factors would go far beyond the accuracy of the data, or the patience of the investigator. For this reason, in this forthcoming final paper a method is outlined whereby a company may determine experimentally the time lag, or an equivalent factor, associated with its given burner face and a given fuel, as a function of firing rate and mixture ratio.

As indicated already, I agree with the discussers that there is no pressing need for further fundamental studies on multiple-port gas-fired units. The time has come for the individual manufacturers to learn more about their particular units, from the point of view outlined in the paper. After sufficient data are obtained, it might be the subject of an interesting paper to review and analyze the results for a large number of burner-face designs.

1642

No. 1642

EVALUATION OF AIR CLEANERS FOR AIR CONDITIONING AND VENTILATION

Part I—Apparatus

By K. T. Whitby*, A. B. Algren**, R. C. Jordan† and J. C. Annis‡,
Minneapolis, Minn.

This paper is the result of research sponsored by the American Society of Heating and Air-Conditioning Engineers, in cooperation with the Mechanical Engineering Department, University of Minnesota, Minneapolis, Minn., and the U. S. Public Health Service.

SINCE WORLD WAR II, the number, types and diversity of application of air cleaners for ventilation and air conditioning purposes has grown rapidly. This growth has resulted in a continuing interest on the part of the ASHAE and the air cleaner manufacturing industry to develop air cleaner rating and testing methods which are commensurate with current needs.

In 1954, following the recommendations of the TAC on Air Cleaning, the ASHAE established a cooperative research project at the University of Minnesota directed toward doing such basic research as necessary to develop improved apparatus and procedures for the testing of all types of air cleaners for occupied spaces. The scope of this research was later increased with the receipt by the University of a Public Health Service Research Grant, with somewhat similar objectives.

This paper is Part I of 3 parts describing general purpose apparatus and procedures for the testing of air cleaners. The apparatus described should be considered as tentative since evaluation is still in progress and it has not yet been submitted as a recommendation for a standard to the ASHAE.

Part II will describe the application and evaluation of the apparatus for arrestance testing of air cleaners and Part III will cover its application to the determination of the life characteristics of air cleaners.

Requirements of a General Purpose Test Method: Commercial air cleaners are presently available in types ranging from low priced high-velocity lint filters to

---

* Assistant Professor, Mechanical Engineering Department, University of Minnesota.
** Professor, Mechanical Engineering Department, University of Minnesota. Member of ASHAE.
† Professor and Head, Mechanical Engineering Department, University of Minnesota. Member of ASHAE.
‡ Research Fellow, Mechanical Engineering Department, University of Minnesota.
Presented at the Semi-Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Minneapolis, June 1958.

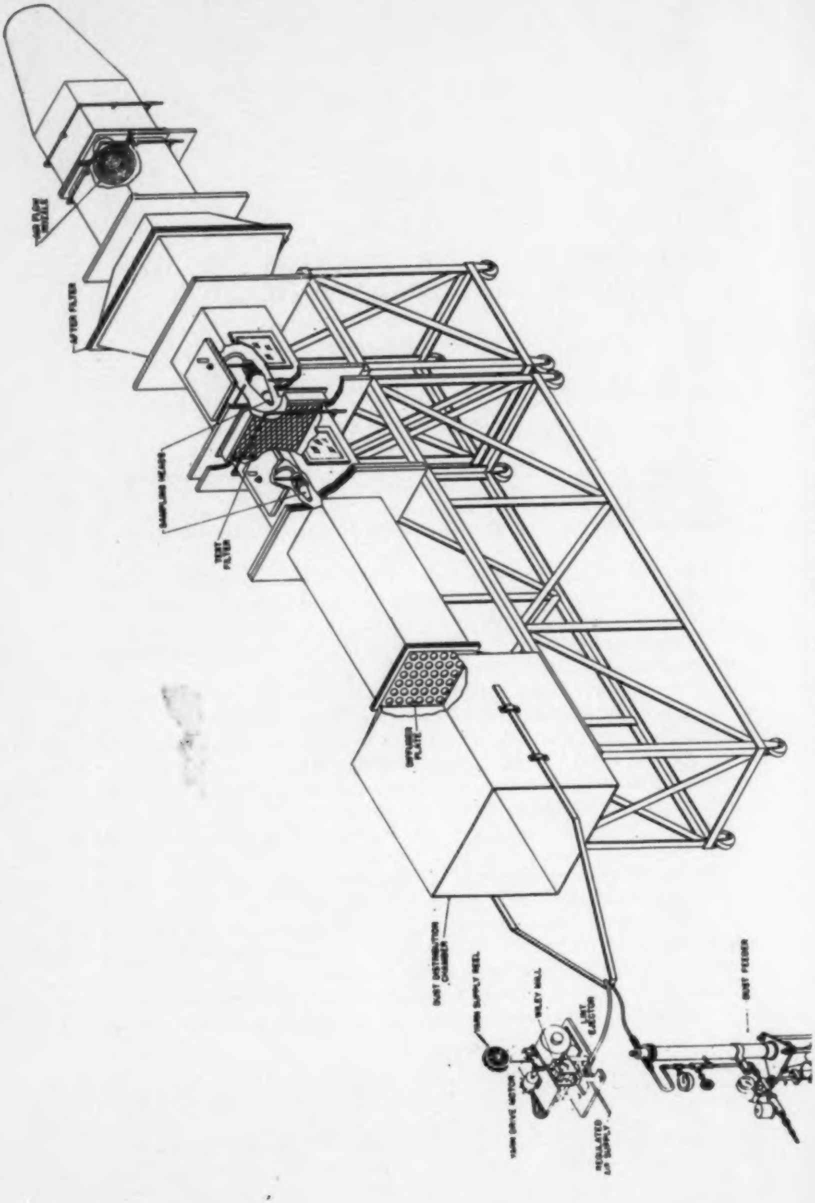


FIG. 1a—COMPLETE AIR CLEANER TESTING APPARATUS

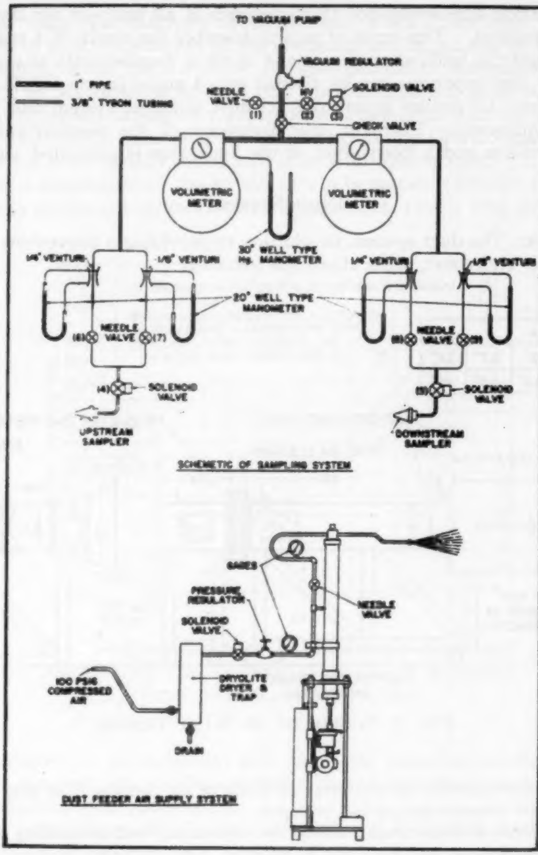


FIG. 1b—DIAGRAM OF SAMPLING SYSTEM (TOP) AND DUST FEEDER AIR SUPPLY SYSTEM OF AIR CLEANER TESTING APPARATUS

higher priced *absolute* type filters for critical applications. This broad span of performance and operating characteristics requires that any method of air cleaner testing intended for universal acceptance must have an unusual degree of versatility. Furthermore it must be inexpensive and simple enough to operate that these factors will not deter potential users.

Extensive consultation early in the project with those experienced in air cleaning indicated that the 2 improvements most needed were: (*a*) an aerosol-generator combination that would better reproduce airborne dust characteristics; and (*b*) a versatile aerosol sampling system that would permit evaluation of dust spot,

weight or particle size arrestance characteristics of all kinds of air cleaners with one set of apparatus. This series of papers describes the results of 4 years of work directed toward the fulfillment of as many of these requirements as possible.

Air cleaner test apparatus can be divided into 4 major functional divisions *viz.*: (*a*) duct system, (*b*) aerosol generators, (*c*) dust sampling system and (*d*) sample evaluation equipment. Therefore the discussion of the requirements of each functional division and a description of the apparatus is presented separately.

DUCT SYSTEM

Requirements: The duct system, in addition to providing a convenient mount for the air cleaner under test, must afford the following:

DUCT SIZE	A	B	C
12"	12"	3.7"	36"
20"	18.5"	5.6"	46"

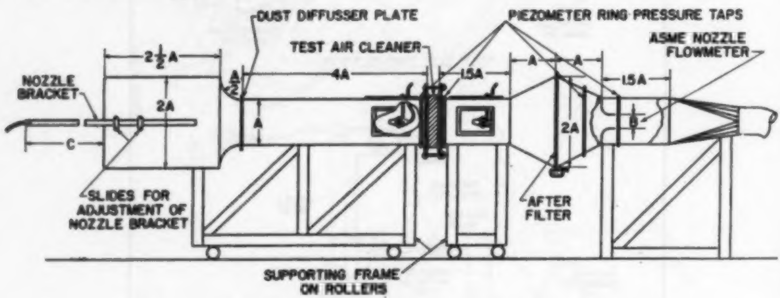


FIG. 2—SCHEMATIC OF WIND TUNNEL

1. a convenient method for mounting all kinds of air cleaners from thin filter media to electronic air cleaners several feet in depth;
2. a convenient and accurate method for measuring and controlling the air flow through the cleaner;
3. satisfactory distribution of the aerosol over the cross section of the tunnel at the test station;
4. a flat velocity profile at the test station;
5. no significant changes in aerosol composition between entrance and test station due to settling in the tunnel;
6. compactness, to save laboratory space.

Fig. 1*a* illustrates the air cleaner test apparatus and Fig. 1*b* shows some of the components diagrammatically. Fig. 2 includes the important dimensions of the duct system.

The air cleaner apparatus operates as follows: The high-velocity jets of dust and air from the dust feeders cause large scale turbulence and thorough mixing in the distribution chamber; the perforated dust diffusion plate at the duct entrance breaks up the large-scale turbulence and distributes the air and dust so that a flat velocity profile and a uniform dust distribution exist at the test station.

The after filter of a relatively effective medium has 2 functions. It serves to clean the air of dust not removed by the air cleaner and it can also be used for whole stream sampling as in the *Air Filter Institute* method of determining weight arrestance. The flow nozzle is so sized that when used with a 14-in. combined draft gage and manometer, satisfactory accuracy is obtained for duct velocities ranging from about 150 to 900 fpm. For maximum flexibility a blower capable of 1 in. of Hg at 900 fpm duct velocity is required if an after filter is used.

Duct size is determined by the air cleaners to be tested. Though the University of Minnesota equipment utilizes a 12-in. square duct, 18½ x 18½ and 24- x 36-in.

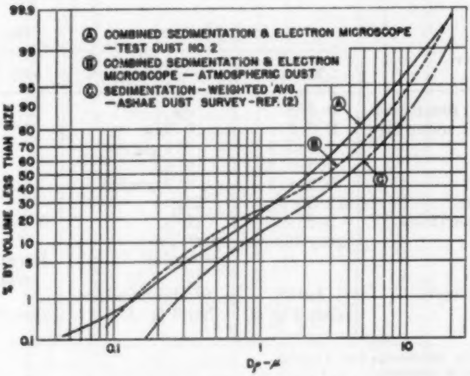


FIG. 3—PARTICLE SIZE BY COMBINED ELECTRON MICROSCOPE AND SEDIMENTATION ON AN AIRBORNE DUST AND NO. 2 TEST DUST

ducts constructed by an industrial firm have also operated satisfactorily. The largest duct required some additional baffles in the dust distribution chamber to obtain satisfactory dust distribution.

Though a vertical duct will eliminate all settling of the test dust, this position is impractical in most laboratories. Settling of the proposed test dust in the horizontal duct is negligible and that of the lint is less than about 5 percent for duct velocities above 300 fpm. For applications requiring lower duct velocities the lint feeder discharge can be moved closer to the test station.

AEROSOL AND AEROSOL GENERATORS

The many aerosols and aerosol generators that have been used for evaluating air cleaners have been reviewed in detail in a previous reference.[1] Some of the aerosols that have been used for the testing of ventilation type air cleaners are natural airborne dust, standardized fine Arizona road dust, Cottrell precipitate, carbon black, mixtures of the foregoing with lint such as the *AFI* test dust and certain oil smokes such as those of dioctylphthalate.

---

[1] Exponent numerals refer to References.

Aerosol or Dust Requirements: Studies of a number of those dusts and of the properties of natural airborne dust which determine air cleaner performance established the following desirable properties for a general purpose test aerosol.

1. Physical properties, such as density and particle-size distribution, should approximate those of average airborne dust. Properties of average air-borne dust as determined from the ASHAE airborne dust survey[3], and the properties of the proposed No. 2 Test Dust are given in Table 1.

TABLE 1—PHYSICAL PROPERTIES OF AVERAGE AIRBORNE DUST AND OF PROPOSED TEST DUST

PROPERTY	AIRBORNE DUST		NO. 2 TEST DUST
	RANGE	AVERAGE	80% COAL, 20% K-1
Volume geometric mean size	0.8–8	3μ	2.2[a]
Number geometric mean size	0–0.2	0.03μ	0.03[a]
Particle density	1.5–3.0	2.1 gm/cc	1.70
Porosity	0.5–0.85	0.65	0.6
Bulk density	0.2–1.5	0.7 gm/cc	0.5
Upper particle size	15–50	25μ	25
Fraction fibrous particles	3–35	10%	0 to 100% possible
Color	gray-black	dark gray	black
Shape		Irregular	Irregular + carbon chains
Electrical Resistivity[b]	1.5×10^8	10^8 ohms cm^2/cm	54[c]
Size distribution	(See Fig. 3)	(See Fig. 3)	(See Fig. 3)[a]

[a] After dispersion by Minnesota No. 3 feeder.
[b] Under 130 kg per cc pressure.
[c] Resistivity of K-1 carbon alone 1.5, of coal alone 7000.

2. Arrestance and loading characteristics of representative types of air cleaners on the test dust should be approximately equivalent to those on average airborne dust.
3. Production, storage and dispersion problems must be reasonable. Use of a single base dust for dust spot and weight arrestance tests and for loading tests is desirable.
4. Base dust and lint aerosol generators should be separate for maximum flexibility of operation and to eliminate the problem of blending and dispersing a mixture of lint and base dust.

Base Dust: After considerable research and investigation most of the requirements just listed were met by a mixture of 80 percent anthracite coal and 20 percent K-1 carbon, hereafter designated as No. 2 Test Dust, prepared in the following manner:

Five hundred fifty (550) grams of oven-dried high density anthracite coal (ρ = 1.7 gm per cc) is ball milled at 67 rpm in a one gallon porcelain jar with 2.6 kg of ¾ to 1½ in. flint pebbles for 72 hr. One hundred thirty eight (138) grams of oven dried K-1 carbon is then added and the mixture ball milled for 8 hr. The prepared dust is then stored in air tight containers in the presence of silica gel until used.

A comparison of the most important physical properties of average airborne dust and the proposed No. 2 Test Dust is given in Table 1. It will be noted that this dust approximates the density, particle size distribution and shape of average air borne dust quite well. Though its electrical resistivity is low, considerable experience in using the dust for testing electric air cleaners has indicated no adverse effects.

TABLE 2—REPRODUCIBILITY OF NO. 2 TEST DUST PENETRATIONS ON 3 REPRESENTATIVE AIR CLEANERS BY DUST SPOT AND WEIGHT

AIR CLEANER	PERMANENT D_{fe} = 400; S_{fe} = 8[c]				ELECTRONIC				20 OZ FELT, WITH D_{fe} = 6.9; S_{fe} = 39							
PENETRATION[b]—%	P_w[e]				P_d				P_w				P_d			
DUST LOT NO.	6	9	17	A[a]	6	13	17	A[a]	6	13	17	A[a]	6	13	17	A[a]
Run 1	62.2	60.6	60.4	81.3	8.62	9.18	9.12	10.90	32.0	32.2	30.4	58.4	62.5	67.8	70.3	77.4
Run 2	62.3	59.6	61.5	60.5	8.64	9.25	9.12	10.70	31.9	32.9	29.3	39.2	65.8	65.7	69.1	81.8
Run 3	60.2	59.6	62.4	72.2	8.69	9.28	9.12	10.52	32.3	32.6	30.8	68.0	66.1	67.3	64.3	82.0
Run 4												38.7				78.0
Mean	61.9			71.3	9.00			10.71	31.9			51.1	66.5			79.8
F	0.84				223				18.7				1.48			
$F_{0.05}$	5.14				5.14				5.14				5.14			
S (Dust Lot)—%[d]	Not Significant				4.2				4.7				Not Significant			
S (Exp.)—%[d]	1.5			12.0	0.14			0.9	1.6			24.6	3.4			2.7
$P_A/P_{No.\ 2\ Test\ Dust}$	1.15				1.19				1.60				1.20			

[a] Normal airborne dust in the Particle Technology Laboratory.
[b] Arrestance or efficiency = 100 − Penetration.
[c] The effective fiber size, D_{fe}, and the filter solidarity factor S_{fe} have been determined from filter pressure drop *vs.* velocity data by a procedure which will be described in a future paper.
[d] Variances given are percent of the penetration value.
[e] No adhesive.

Table 2 illustrates the reproducibility of dust penetration measurements using the No. 2 Test Dust and normal airborne dust on 3 representative types of air cleaners. The permanent filter was washed with benzol after each run. No adhesive was used. Later papers will include data on the effect of adhesives. Face velocity was 520 fpm.

The electronic air cleaner was of the conventional 2-stage ionizer and plate type. Good voltage regulation and measurement was necessary to achieve this degree of reproducibility.

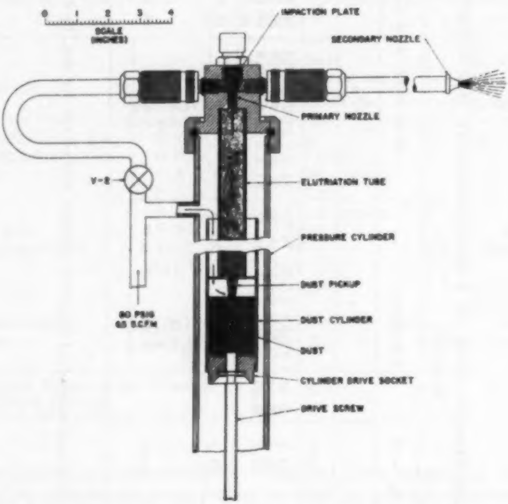


Fig. 4a—Dust Feeder—Schematic of Feeder Head

The 20-oz wool felt was untreated and was run at 42 fpm face velocity using the filter media attachment shown in Fig. 6. A new sample of felt was used for each run on each dust.

From Table 2 it may be concluded that variability in the air cleaner being tested and in the different dust lots is greater than the variability of the evaluation techniques used. The high experimental variance S for the felt on laboratory airborne dust is due to the large variation in relative proportion of the larger particles in airborne dust as determined by the ASHAE airborne dust survey. It will be noted that the ratio of the penetration on laboratory airborne dust to the penetration on the No. 2 Test Dust is greater than one for these experiments. From the previous dust survey[2] it is known that the airborne dust in the particle laboratory is significantly finer than the average shown by the dust survey. Probably P_d and P_w on airborne dust in other industrial cities will be found to be more nearly equivalent to those of the No. 2 Test Dust.

In Part II, comparative data on several other test dusts and dust feeders will be presented, showing that the reproducibility of the new sonic jet feeder Minnesota

No. 3, No. 2 Test Dust combination is equal to or better than other combinations.

Base or Aerosol Dust Generator: Research on the dispersibility of fine dusts having size distributions similar to airborne dust showed that ordinary aspirator type aerosol generators were incapable of dispersing such dusts reproducibly and completely. To obtain the better dispersion deemed necessary, the sonic jet dust feeder, Minnesota No. 3, illustrated in Figs. *1a*, *4a*, and *4b* was developed, incorporating the following features:

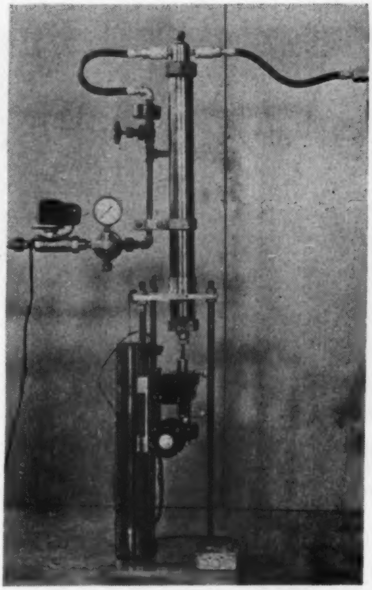


FIG. 4b—DUST FEEDER—DUST FEEDER ASSEMBLY

1. Good dispersion of the dust as a result of the passage of the air-dust mixture through 2 critical nozzles in series and the impaction of the dust at near sonic velocities on an impaction plate;
2. The second critical orifice mounted at the point of dust discharge to eliminate the effects of agglomeration in the discharge hose;
3. Feed rate variable from 0.05 to 10 gms per min, achieved through the use of interchangable dust pickups and dust cylinders in combination with a variable speed drive;
4. Elutriator tube to reject large dust aggregates.

Operation: The feeder head and attached dust pickup are removed by unscrewing the top cap. This permits removal of the dust cylinder for loading with dust and weighing during filter loading procedures. In order to permit the feeding of a predictable weight of dust per unit time, the dust may be compacted to a known bulk density within the dust cylinder. As long as both the dust and compressed air are dry, compacting the dust does not seem to affect the dispersion.

Regulated dry compressed air at approximately 0 F dew-point and 80 psig enters the pressure chamber and flows down between the elutriator tube and dust cylinder to the pickup foot. Here the thin slice of dust scraped off by the rotation of the dust against the pickup foot is carried through the foot slot, up the elutriator tube, through the primary nozzle, against the impaction plate and out the discharge hose to the secondary nozzle. To maintain the desired pressure ratio across the secondary nozzle as the nozzles wear, a small amount of air may be bled through

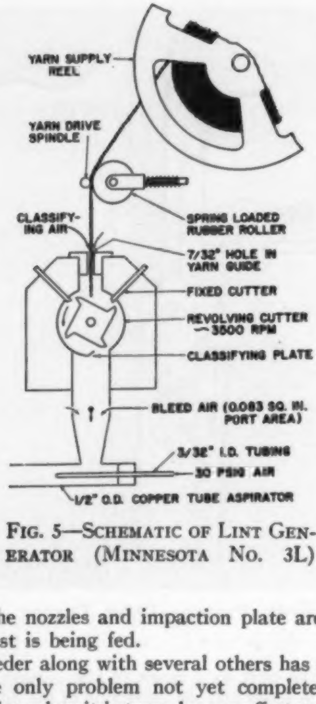


Fig. 5—Schematic of Lint Generator (Minnesota No. 3L)

valve, V-2 Fig. 4*a*. The nozzles and impaction plate are replaceable since they wear when abrasive dust is being fed.

Evaluation of this feeder along with several others has shown it to be versatile and reproducible. The only problem not yet completely solved is occasional puffing and uneven feeding when it is turned on or off at very low feed rates. This occasionally creates a problem during dust spot evaluations where very low optical densities are required.

Lint Generator: The life and hence the maintenance cost of ventilation type air cleaners depends primarily on the concentration of fibrous particles in the air being cleaned. Therefore any good test procedure must satisfactorily reproduce the fibrous fraction.

Studies* have shown that it is difficult to blend and feed mixtures of base dust and a lint having a character and concentration similar to that of normal airborne dust.[2] Therefore the separate lint generator shown in Figs. 1*a* and 5 has been

* The filter loading portion of the ASHAE Dust Survey will be published in the future.

developed from the standard medium size mill. This generator is capable of continuously converting fine cotton knitting yarn into a lint aerosol that is a reasonable approximation of natural lint. The problem of generating a broad distribution of fiber lengths has been solved by the combination of the right amount of classifying air leaking through the yarn guide and a ⅛-in. *ski jump* at the trailing edge of the classifying plate. The *ski jump* on the classifying plate causes some of the fibers to make several revolutions within the mill, thereby being reduced in length, while others pass through with lengths up to ¼-in. or more. Different size yarn drive spindles are used to obtain different feed rates. A ¼-in. diameter

A—sampler body, B—sampling media, C—retaining ring, D—sampler front, E—high performance filter mat attachment, F—test filter mat, and G—isokinetic tip

FIG. 6—EXPLODED VIEW OF SAMPLING HEAD AND HIGH PERFORMANCE FILTER MEDIA ATTACHMENT

spindle driven at 15 rpm will feed about 0.25 gms per min. Comparisons of the performance of representative air cleaners loaded with natural airborne dust and with No. 2 Test Dust—lint mixtures will be presented in Part III.

SAMPLING SYSTEM

Function: The sampling system is used to obtain a representative sample of the dust upstream and downstream of the test air cleaner. The basic requirements for a satisfactory sampling system can best be understood by considering the permissible range of variation of the variables for the calculation of air cleaner performance in Equation 1.

$$P = \frac{C_d A_d / q_d t_d}{C_u A_u / q_u t_u} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

For practical reasons A_u is usually made equal to A_d. Since the amount penetrating an air cleaner may vary from 100 percent to less than 1 percent this means that provision must be made for accurate measurement of the ratio $(C_d\ q_u\ t_u)/(C_u\ q_d\ t_d)$ over nearly a 100 to 1 range. Such common concentration measuring

methods as the dust spot and weight method are limited to approximately a 5 to 1 range or less where C is proportional to the measurement made. Thus, it is necessary to vary the total quantity of air sampled, Q_u, upstream. This may be done by varying either q or t. Practical considerations make it difficult to use more than about a 20 to 1 ratio of q_d/q_u, so that variable time sampling must also be used for penetration measurements of less than 1 percent.

A thorough theoretical and experimental study of all of these factors has led to the development of the system to be described. It has been designed to provide flexibility and accuracy over a broad range of conditions. Also considerable attention has been given to convenient and reliable operation.

TABLE 3—DOWNSTREAM/UPSTREAM AIRFLOW RATIOS FOR DIFFERENT SAMPLER TIP COMBINATIONS

Downstream		Upstream Tip No.							
Tip No.	Dia., Inches	None	1	2	3	4	5	6	7
None	1.375	1	1.21	1.72	2.56	5.15	8.20	15.5	30.3
1	1.25	0.825	1	1.42	2.11	4.25	6.77	12.6	25.0
2	1.05	0.581	0.704	1	1.49	3.00	4.76	8.89	17.6
3	0.860	0.390	0.473	0.670	1	2.01	3.21	5.97	11.8
4	0.607	0.194	0.235	0.333	0.497	1	1.60	2.97	5.90
5	0.481	0.122	0.148	0.210	0.31	0.63	1	1.86	3.70
6	0.352	0.065	0.079	0.112	0.17	0.34	0.54	1	1.98
7	0.250	0.033	0.040	0.057	0.085	0.17	0.27	0.50	1

Tip No.	P_s, Inches Hg.	
	20	15
None	288	340
1	336	410
2	476	580
3	710	867

Maximum duct velocities at which isokinetic sampling can be achieved with various tips using 1106 BH glass fiber paper, fpm.

Features: 1. Sampling heads have been designed to accommodate 47 mm diameter disks of all common sampling media such as W-41 chemical filter paper, glass fiber paper, and millipore filters. This permits microscope count, light transmission measurement, weighing or sedimentation size analyses of the dust sampled.

An exploded view of the sampler with a special high performance media attachment is shown in Fig. 6. The main components of the sampler parts A and D are modified from old style commercially available millipore filter holders. In normal use the isokinetic tips G screw directly into D. The inside diameter of ring C and the active area of the porous stainless steel backing plate in A must be exactly equal to provide a sharply defined spot for dust spot evaluations.

Attachment E is provided to enable the evaluation of 3-in. diameter pieces of high performance filter media at face velocities between 10 and 70 fpm while operating the duct system at velocities high enough to obtain good dust distribution and accurate flow measurement. The isokinetic tips screw into the front part of E.

Seven isokinetic tips having the dimensions given in Table 3 have been used. Tips have a 15 deg taper and 1.375 in.—18 threads. Table 3 also gives the isokinetic upstream to downstream air flow ratios for various combinations of tips. For the testing of specific types of air cleaners, all of these tips would not be necessary.

2. Rather elaborate air flow metering equipment has been found necessary to test all kinds of air cleaners with acceptable accuracy. From Fig. 1b, it will be noted that 2 types of flow meters are provided. Dual range venturi flow meters are provided in each sampling line for accurate setting and measurement of flow rates. Precision variable area rate meters could be used instead of the venturi meters. In addition volumetric meters are provided. It will be noted that $q_u\, t_u$ and $q_d\, t_d$ represent the total volume of air drawn through the upstream and downstream sampling media respectively. Thus the volumetric meters permit direct measurement of these air volumes unaffected by small changes in airflow rate due to loading of the sampling media. Also short interval variable time sampling can be incorporated without the use of accurate time measurement.

The calibration accuracy of the metering system depends on the accuracy required. All meters used in the University of Minnesota test setup have been calibrated to better than ±1 percent. For most air cleaner testing ±2 percent is probably adequate.

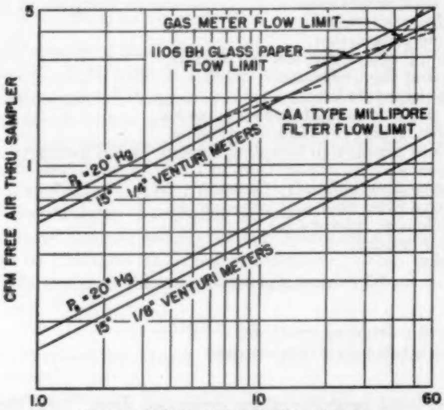


FIG. 7—AIR FLOW RELATIONS FOR SAMPLING SYSTEM VENTURI FLOW METERS

Fig. 7 gives the flow meter calibrations for the venturi meters. Accurate measurement can be obtained from about 0.2 to 4.0 cfm of free air. Also shown are the gas meter limits and the maximum air flows which can be obtained using type AA millipore filters and 1106 BH glass fiber paper. The latter appears to be best for most purposes.

The upstream and downstream flow meters are connected to the vacuum system so that they always operate at the same pressure. This permits using the ratio of the indicated airflows without correction for pressure in Equation 1.

3. This sampling system may be used to test air cleaners with arrestances above 99 percent by cycling the upstream solenoid. Since the volumetric meters integrate the actual air flow, accurate measurement of the on-off periods is not required. To provide a statistically valid sample, however, the rate of cycling should be such that at least 20 samples are taken during a run. The timer used permits adjustment of the on-period from 1 to 60 sec and the off-period from 5 sec to 5 min.

4. From Fig. 1*b* it will be noted that a bleed needle valve (2) and solenoid valve (3) in series have been provided so that when the sampling system solenoid valves are closed, the valve (3) opens thereby keeping the air flow through the vacuum pump unchanged and the static pressure in the meters the same. This permits instantaneous starting and stopping of the sampling system without any gas meter integration errors due to pressure changes. Though not necessary, a vacuum regulator makes for more convenient operation of the system.

Control of the entire air cleaner system is possible by either an interval timer or a special dust control device which are provided but not illustrated here. The control device which will be described in a later paper, samples the aerosol in parallel with the upstream sampler and continuously evaluates optically the dust deposited on a piece of filter medium. This instrument can be set to shut off any part or all of the entire air cleaner apparatus after a predetermined amount of dust is fed. It is particularly useful when running dust spot evaluations with either atmospheric or No. 2 Test Dust.

Operation: Because of the many modes of operation required for the evaluation of different types of air cleaners by the different methods, such as dust spot and

NOMENCLATURE

P = penetration (arrestance = $1 - P$).
P_d = dust spot penetration.
P_w = weight penetration.
P_v = volume penetration.
q = sampling rate, cubic feet per minute.
t = sampling time, minutes.
A = sampling media area.
C = dust per unit area of sampling media.
ρ = density, grams per cubic centimeter.
P_s = absolute pressure in sampling system, inches mercury.
S = variance.
S_{fe} = effective filter solidarity factor.
D_{fe} = effective filter fiber size, microns (μ).
D_p = particle size, microns (μ).
OD = optical density.
F = F ratio.
$F_{0.95}$ = F ratio for 95 percent significance level.

Subscripts

u = subscript denoting upstream.
d = subscript denoting downstream.

weight, only the general procedures are described here. Specific procedures will be discussed in Parts II and III.

In general, an arrestance evaluation requires the 4 following steps.

1. Selection of duct velocity, isokinetic tips, and sampling rates with the aid of Table 3 and graphs similar to Fig. 7, so that the amount of dust collected upstream and downstream will be roughly the same.

If the high performance filter media attachment shown in Fig. 6 is used, the downstream sampling rate is determined by the face velocity at which it is desired to test the media. As illustrated, the attachment is designed to test 3-in. active diameter media specimens. This is satisfactory for face velocities down to about 10 fpm. Lower velocities would require a larger active area to maintain adequate downstream sampling rates.

2. Adjusting of the sampling system with clean, previously evaluated sampling media installed, by means of the appropriate valves.

3. Taking of appropriate size dust samples (0.1 to 0.9 O.D. for dust spot and 5 to 15 mg for weight) with the dust feeder operating at an appropriate feed rate. Dust spot runs may be as short as 1 min, while weight evaluations using natural atmospheric dust as the test aerosol may require 24 hr or more.

4. Removal of sampling media and evaluation by the appropriate method, (*i.e.* particle count, light transmission, weight or sedimentation size analysis).

The air cleaner penetration is then calculated from Equation 1.

Simulated air cleaner life or loading procedures are still being evaluated so that

the procedure to be described here is tentative. In general this procedure is required.

Dust feeder, lint generator and sampling system are prepared and adjusted to the desired rates. Dust and lint are then fed and the dust spot or weight penetration and the dust cylinder and yarn reel weights are then determined at preselected pressure drops (*i.e.* clean, 0.2-, 0.35- and 0.5-in. water gage for a typical panel filter).

Weight penetrations may be determined with either the sampling system or from the weights of the test filter, after filter and dust cylinder.

SAMPLE EVALUATION

Weight: An ordinary 0.1 mg sensitivity analytical balance is satisfactory for weighing the 1106 BH glass fiber paper used for weight penetration measurements. This paper has no binder and shows negligible weight changes with normal humidity variations.

Dust Spot: A photometer of more than ordinary sensitivity is required to measure the light transmission of the 1106 BH paper accurately. A null photometer such as that described previously[3] is most satisfactory. W-41 chemical paper is not recommended because of its low arrestance (85 percent) on the No. 2 Test Dust.

Particle Count: Count arrestance is readily determined from particle counts on millipore filters. Either light or electron microscope counts may be used.

Sedimentation Size Analysis: Centrifuge sedimentation particle size analysis of samples collected on millipore filters can be made by previously described techniques[2]. The sediment column height for the upstream and downstream samples may also be used to calculate the dust penetration. This volume penetration, P_V, is approximately equal to the weight penetration.

REFERENCES

1. *Air Pollution Handbook*, by Magill, Holden and Ackley (McGraw-Hill Book Co., Inc., New York, 1956).

2. ASHAE RESEARCH REPORT No. 1627—The ASHAE Air-Borne Dust Survey, by K. T. Whitby, A. B. Algren, R. C. Jordan and J. C. Annis (ASHAE TRANSACTIONS, Vol. 64, 1958, p. 129).

3. ASHAE RESEARCH REPORT No. 1597—The Dust Spot Method for Evaluating Air Cleaners, by K. T. Whitby, A. B. Algren and R. C. Jordan (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 171).

DISCUSSION

R. D. RIVERS*, Louisville, Ky. (WRITTEN): The authors have already earned a reputation in the air filtration field for their basic studies. The present paper continues this work, and makes new contributions to the filter testing art. The authors have recognized the problems involved in dispersing a test dust which will approximate airborne dust, and they have attacked these problems with vigor.

Separating lint and dust feeders seems a wise arrangement, and one which will provide a more uniform feed for both constituents. The lint generator with its *ski jump* to provide a range of fiber lengths, is clever. The dust feeder seems an improvement over the usual low-velocity aspirator type feeder, and we await with interest the publication of Part II of the authors' paper describing this in detail. Wright† in 1950 described an

---

* American Air Filter Company.

† A New Dust-Feed Mechanism, by B. M. Wright (*Journal of Scientific Instruments*, Vol. 27, No. 1, January 1950, p 12) Also described in *Particle Clouds*, by Green and Lane (p. 53, published by E & F N. Spon, Ltd., London, 1957).

almost identical feeder which was highly successful, and which is thought to be commercially available in England. This feeder did not include an elutriation chamber as in the Minnesota #3, however. Such a chamber should be desirable.

Table 2 is very interesting, but seems to require further explanation and careful checking by the authors of the paper. Accepting the data listed above the line labeled "Mean", fourteen errors are to be found in the calculation below the line.

The discussion which follows assumes that by the term *variance—S (Dust lot)—%* and *S (Exp.)—%* the authors actually mean standard deviations. (It is somewhat illogical to express a *variance* as a percent of mean, for they are not dimensionally the same; several statisticians have informed us that they have never seen the variance so expressed. If we do express it so, the correct value of variance for Column 4 under the permanent filter, becomes 153 percent of the mean—a difficult number to comprehend.) For purposes of discussion, a corrected table of values is appended as Table A, with percent standard deviation tabulated. It must be noted, however, that the limited number of samples—particularly on atmospheric dust—would suggest some caution in interpretation.

The discusser has assumed that the term *S (Dust lot)—%* is meant to indicate the variance of the means of the 3 runs on each lot rather than the mean of the variances. In addition, the term *S (Exp)—%* is assumed to mean the variance among all 9 runs for each filter.

The *F-test* correctly calculated shows that we may rely on the fed-dust test procedure for the permanent filter and the 20 oz felt filter. For the electronic filter, something happened to make dust lot 6 yield a significantly different penetration. This may seem odd, in that the standard deviation of this group of 9 measurements is one of the smallest in the table. The explanation is, perhaps, that the combination of filter and test method has its highest stability and resolution under these conditions, and so the data distinguish between two slightly different conditions. Note that under this condition—electronic filter with dust spot test—the use of atmospheric dust is also highly consistent. The same is true of the 20 oz felt, *tested by dust spot.*

In short, the data indicate that atmospheric dust is satisfactory for dust spot and highly unsatisfactory for weight testing. Accuracy is not the only factor, of course; as the authors state, weight evaluations using atmospheric dust require 24 hours or more—an inconveniently long test. This agrees with the evidence in their paper (Ref. 2) which would indicate a maximum target load of 15 mg in 24 hours using the maximum flow of which the samplers are capable.

An item of considerable interest in evaluating the test procedure would be data on its performance when *no* filter is in place. The data in the paper mingle the effect of the test procedure and the filter performance in such a way that they cannot be separated.

The samplers are neat and have considerable flexibility, but one wonders about the sealing between lucite ring C and target "B" in Fig. 6. The authors laid considerable stress on the need for good sealing in ASHAE Research Report 1597 (Ref. 3). The seal in Fig. 6 seems identical with the regular millipore sampler. Have improvements been made here?

The air flow metering system is indeed elaborate, and probably necessary if the equipment is to be used *to test all kinds of air cleaners with acceptable accuracy.*

Statements made in the paper imply that the system is useful for efficiencies as high as 99 percent. The writer doubts that a test stand where dusts are fed, even occasionally, at 1000 times normal airborne concentrations, can be used to test 99 percent filters. Even the small high-performance filter media attachment would be better off outside such a duct. Experience in filter testing has shown time and again that filters fall into distinct classes each one of which requires distinct filter testing methods. The present paper seems to add further evidence to support this concept.

C. B. ROWE, Madison, Wis. (WRITTEN): It is evident that Dr. Whitby and his associates have been working diligently on the project. In going through the paper, my

TABLE A—RECALCULATION OF DATA OF TABLE 2

AIR CLEANER	PERMANENT				ELECTRONIC				20 OZ FELT							
PENETRATION (%)	(WEIGHT TEST)				(DUST SPOT TEST)				(WEIGHT)				(DUST SPOT)			
DUST LOT NO.	6	9	17	ATM.	6	13	17	ATM.	6	13	17	ATM.	6	13	17	ATM.
Run 1	62.2	60.6	60.4	81.3	8.62	9.18	9.12	10.90	32.0	32.2	30.4	58.4	62.5	67.8	70.3	77.4
Run 2	62.3	59.6	61.5	60.5	8.64	9.25	9.12	10.70	31.9	32.9	29.3	39.2	65.8	65.7	69.1	81.8
Run 3	60.2	59.6	62.4	72.2	8.69	9.28	9.12	10.52	32.3	32.6	30.8	68.0	66.1	67.3	64.3	82.0
Run 4												38.7				78.0
Mean (Dust Lot)	61.6	59.9	61.4	71.3	8.65	9.24	9.12	10.71	32.1	32.6	30.2	51.1	64.8	66.9	67.9	79.8
Mean (All Lots)		61.0*				9.00				31.6*				66.5		
F		2.70*				220.				3.12*				1.48		
$F_{0.05}$		5.14				5.14				5.14				5.14		
**S (Dust Lot) %		1.5				3.6*				4.0*				2.4		
**S (Exp) %		1.9*		14.7*		3.0*		1.81*		3.7*		28.4*		3.6		3.1*
P_A/P No. 2 Dust		1.17*				1.19				1.62				1.20		

** These two lines are redefined to mean *percent standard deviation*.
* Denotes calculated values which differ significantly from those in Table 2.

basic reaction is a hope that the equipment and procedures can be considerably simplified before they are considered as final. The cost and the complexity of the equipment in its present state would put the testing of filters in the category of that which would be indulged in only by those who have a very major interest in the field. In fact, there has been some resistance to the cost and complexity of the *AFI* Code equipment which is considerably simpler and less expensive than that which the authors propose.

I note, too, that the equipment involves details which have caused controversy over the past several years. That is, the use of partial samplers with the resulting controversies of isokinetic sampling, uniformity of dust distribution on the test duct, and uniformity of velocity distribution in the test duct.

R. J. VAN DOORNEVELT, Philadelphia, Penna.: I have noted with interest that in describing his apparatus, techniques and flexibility the author made no mention of

TABLE B—RECALCULATION OF STATISTICAL DATA BY UNIVERSITY OF MINNESOTA FOR TABLE 2

	PERMANENT		ELECTRONIC		FELT, BY WEIGHT		FELT, BY DUST SPOT	
	No. 2	ATM.	No. 2	ATM.	No. 2	ATM.	No. 2	ATM.
Mean (All Dust Lots)	61.0[a]	71.3	9.00	10.71	31.6[b]	51.1	66.5	79.8
F	2.70[b]		223		18.7[c]		1.48	
$F_{0.05}$	5.14		5.14		5.14		5.14	
Dust Lot % Std. Deviation	2.58[c]		4.2[c]		4.7[c]		1.7[c]	
Experimental % Std. Deviation.	1.5[c]	12.0[c]	0.40[c, d]	1.44[c, d]	1.6[c]	24.6[c]	3.4	2.7[c]
$P_A/P_{\text{No. 2 Test Dust}}$	1.15		1.19		1.60		1.20	

[a] Typographical error by University of Minnesota.
[b] Errors made by University of Minnesota in original Table.
[c] Errors by discusser in his recalculation of Table 2.
[d] Differences due to number of digits carried in calculations by University of Minnesota.

testing or precedures to determine air filter efficiency on viable airborne micro-organisms. I would like to know if this is being considered, and also if the proposed sampling method is considered practical for this type of work.

J. R. SWANTON, JR., Cambridge, Mass. (WRITTEN): I recognize the difficulty in creating any test dust that will bear any acceptable relationship with natural airborne dust. There is not just the problem of making the test dust, but the inherent impossibility of generalizing on the properties of natural airborne dust. Since it is a variable in itself there can never be any agreement on what is meant by it. The authors have probably come as near as anyone can be expected to do in working out a test dust and setting up a system for handling it which gives results similar to natural dust situations.

More precise data could be obtained on the performance of an air cleaner by running a series of penetration tests covering a range of particle sizes, each test using a single particle size. The result of these tests would be a curve of penetration *vs* particle size and this data should be most informative. The problem is to generate precisely the particle size necessary for each of these tests, but it looks as though this can now be done, at least for a sufficiently representative number of sizes. The penetration measurement itself is a test procedure that can be carried out rapidly for each size.

Lint presents a problem that must be faced in most practical applications of air cleaning equipment. However, if its presence helps the filtering operation, as it is likely to do, this does not indicate a quality of the filter, but rather the fortunate accident of the occasion of lint.

One comparatively minor question about technique: Shouldn't the upstream nozzle be protected against the intrusion of large particles during periods of intermittent sampling?

As a final summary comment, it would seem that if one accepts the test dust, then the procedure appears quite reasonable.

AUTHORS' CLOSURE (Dr. Whitby): The comments of Mr. Rivers are particularly interesting in view of the well-known contributions of his company to the air cleaning field.

The dust feeder mentioned by him is very similar to the University of Minnesota Feeder. However, we feel that the University of Minnesota Feeder is mechanically superior for the purpose intended.

TABLE C—CHECKS WITH NO AIR CLEANER IN THE DUCT

DATE	DUST SPOT PENETRATION, PERCENT	WEIGHT PENETRATION, PERCENT
2/22/58	100.7	
"	100.6	
"	100.0	
"	99.7	
12/31/57	100.4	99.0
"	98.0	103.0
"	100.0	
8/9/57	99.8	
"	100.1	
8/13/57	100.1	101.7
"	99.6	101.1
"	101.3	
"	99.2	
Mean	99.96	101.20
St. Deviation—%	0.77	1.42

Mr. Rivers surmised correctly that the *S* symbol in Table 2 could have been more properly called standard deviation. The values tabulated are the square roots of the mean value of the squares of the deviations about the means in per cent; as calculated by the usual analysis of variance procedure.

All of the calculations in Table 2 have been carefully rechecked using a desk calculator (Table B) and carrying enough digits so that rounding errors should be very small. Of the 14 errors cited by Mr. Rivers, the scoreboard reads as follows: 1—typographical error; 2—University of Minnesota errors—neither changes the significance of the data; 3—difference due to number of digits carried on calculator; nine errors in the discusser's calculations; and three additional errors in the table submitted by the discusser but not claimed.

Mr. Rivers' conclusions regarding the differences between dust lots as indicated by the electronic air cleaner seem to be based on an incorrect interpretation of the column and residual Std. Deviation, see (Dust Lot) and (Exp.) in Table 2.

If the F ratio is larger, this can be due either to a large S (Dust Lot) or a small S (Exp.). S(Exp.) (residual) is the mean of the S about the mean of each column. Thus, in this case the very small S(Exp.) gives a highly significant lot effect even though the S (Dust Lot) is on the same order of magnitude as the S (Dust Lot) for the other air cleaners.

The authors would hesitate to conclude on the basis of the data shown here that the use of atmospheric dust for dust spot is highly consistent. They, as well as other investigators, have too much data to the contrary. More data on this will be presented later.

The authors have done considerable checking of the apparatus with no air cleaner in place. Table C illustrates some representative check runs. Such check runs are made every time the system is changed in any way or a meter is recalibrated. The standard deviation for the 13 runs by dust spot was 0.77 per cent and for 4 runs by weight 1.42 per cent. Dust spot is used more often for checking because it is so much more rapid.

Though the samplers described in the paper do seal satisfactorily, they are not rugged enough, nor is the paper positioning reliable enough for rapid routine work. Since the writing of this paper (Fall 1957) one modification and 2 completely new designs have been tried. The last of these is not only simpler, but appears to meet all requirements very well. Details will be included in a later paper.

The authors are puzzled by the statements in Mr. Rivers' last paragraph. In Part II of this paper, data will be submitted showing that dust spot efficiencies above 99 percent can be measured with useful accuracies. It is not necessary to use dust concentrations 1000 times normal, but on the basis of experience, do not understand why high dust concentrations should have any more adverse effects on high performance air cleaners than high dust concentrations do on the testing of lower performance air cleaners. This later type of testing is universally used on lower performance filters in one form or another by the entire air cleaner industry at present.

The authors also fail to see how the discusser can conclude from this paper that air cleaners fall into distinct classes which must be tested by distinct methods if by methods he means completely different apparatus. The authors believe that in the past separate classes of air cleaners have been created by the existence of separate testing methods rather than the other way around. It has been their aim to develop a single set of apparatus which is capable of satisfactorily evaluating all types of air cleaners within one interpretive scheme so that all air cleaners can be compared to at least some degree on a common basis.

In reply to the question asked by Mr. Van Doornevelt, we have not made any attempts to use the apparatus for determining air cleaning efficiency on viable airborne organisms. This should be possible, however, since the sampling system will take the standard 47 mm diameter millipore filter which has been used for the sampling of such organisms. It should also be possible to replace the filter paper sampling heads with sieve samplers for the sampling of airborne organisms that cannot be sampled satisfactorily on millipore filters.

Mr. Rowe mentioned the subject of the cost of the testing apparatus as described in this paper. Since it is capable of evaluating all types of ventilating air cleaners, the authors believe that the estimated cost of something under $10,000 is commensurate with the function which the apparatus is to perform. For the testing of a specific air cleaner performance range, the apparatus can be simplified somewhat in detail.

While, as Mr. Rowe notes, the use of partial sampling most certainly requires more careful design and operation of the air cleaner testing apparatus, it is impractical to use whole stream sampling and evaluation with the dust spot and particle count methods of sample evaluation under most conditions. Whole stream sampling is practical only for the testing of roughing filters and small pieces of high performance filter media.

The authors agree with Mr. Stanton that the efficiency *versus* particle size characteristic of an air cleaner is the best way of specifying its effectiveness. This is so because, ideally, this characteristic is independent of the aerosol and aerosol evaluation method used.

The chief obstacle to the greater utilization of this method of evaluation is the difficulty in obtaining sharply classified dust fractions, or the generation of homogeneous aerosol particles over a sufficient range of particle sizes.

The authors, along with others, are currently working on an improved homogeneous aerosol generator and are investigating improved means for classifying bulk dusts.

During intermittent sampling no detectable amount of particles have been observed entering the upstream sampling tip. This is readily checked by exposing a sampler to the airstream for a period with the sampler shut off.

1643

No. 1643

WINTER INFILTRATION THROUGH SWINGING-DOOR ENTRANCES IN MULTI-STORY BUILDINGS

By T. C. MIN*, AUBURN, ALA.

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Avenue, Clevland 3, Ohio.

AIR INFILTRATION through entrances has been recognized as an important consideration in estimating heating and air-conditioning loads, especially for tall commercial buildings, in which chimney effects are large and heavy traffic prevails. Data presently available on infiltration[1-4] are limited, and the effects of vestibules, fresh air and exhaust fans, traffic rate, and human obstruction have not been fully evaluated. At the ASHAE Research Laboratory, work on entrance infiltration has been in progress since January 1956, under the guidance of the Technical Advisory Committee on Heating and Air-Conditioning Loads**. The purpose of the investigation is to provide basic information on air infiltration through entrances of commercial-type multi-story buildings. In this paper, the infiltration through various types of swinging-door entrances under winter heating conditions is reported. Infiltration through swinging-door entrances under summer cooling conditions is being studied, and a project on infiltration through revolving-door entrances is being planned.

METHOD OF APPROACH

Entrance infiltration is caused by a pressure differential across the entrance, resulting from chimney effect, supply and/or exhaust fan operation, and wind pressure. For a given pressure differential, the volume of infiltration air passing

---

* Former Research Engineer, ASHAE Research Laboratory, Cleveland, and now Associate Professor, Mechanical Engineering Department, Alabama Polytechnic Institute. Member of ASHAE.

Exponent numerals refer to References.

** *Personnel*: H. T. Gilkey†, *chairman*; P. H. Yeomans†, *vice chairman*; P. R. Achenbach; R. T. Baum; C. W. Coblentz†; Paul Gayman; R. A. Gonzalez; W. S. Harris†; J. N. Livermore; C. O. Mackey; G. R. Munger; I. A. Naman; T. F. Rockwell; A. M. Simpson†; J. P. Stewart; D. B. Turkington†; A. G. Wilson†; W. R. Yeary. (†Indicates a member of Subcommittee on Infiltration).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

through hinged-door entrances depends upon a number of variables, among which are the following:

1. Type of entrance: (*a*) single-bank doors; (*b*) vestibule.
2. Dimension of doors.
3. Arrangement of doors: (*a*) opposite-swinging; (*b*) parallel-swinging.
4. Direction of swinging of doors: (*a*) all swinging out; (*b*) one swinging in, the other swinging out; (*c*) all swinging in.
5. Depth of vestibule.
6. Size of door cracks.
7. Size and location of marquee.
8. Type of door operators or closers.
9. Traffic rate and pattern.
10. Tightness of other parts of the building.

In view of the many variables involved, the work was divided into 3 phases which were conducted simultaneously. Phase 1 was the development of draft factors* which could be used to predict the pressure differential across the entrance of a building of any height, at any outside design temperature and wind velocity, and for any inside design temperature and blower action. Phase 2 was the determination of flow coefficients* for doors of various types and at any angle of opening. In Phase 3, studies were made to develop a correlation between traffic rate and door position for various types of entrances and vestibule depths.

By combining the information from the 3 phases, the infiltration rate for any particular type of entrance, traffic rate, building height, outside weather conditions and blower effect could be determined.

TEST APPARATUS, PROCEDURE, AND CONDITIONS

Phase 1: Pressure differential measurements were made in 23 building entrances in Cleveland and Pittsburgh during the months of January, February, November, and December of 1956 and January and February of 1957. Information regarding the height and cubical content of the buildings and the capacities of exhaust and fresh air supply fans was furnished by the building managers. Most tests were made during the day time under normal traffic and building operating conditions. For studying the effect of blower action, a few tests were run in the evenings when the exhaust and/or supply fans could be turned on and off as desired.

In determining pressure differentials, the outside pressure was sensed by the total-pressure opening of a Pitot tube located near the building and approximately 7 ft above the sidewalk. In most tests, the Pitot tube was pointed in a direction perpendicular to the building entrance. However, in a few tests it was pointed windward and leeward, and readings were taken at several points between the building and the curb.

In tests in Cleveland buildings, the Pitot tube was connected to a pressure transducer and an electronic recorder located inside the building entrance. In the other tests, a draft gage was substituted for portability. All of these instruments were calibrated at the Laboratory prior to their use.

The rubber tubing between the Pitot tube and the pressure meter was passed through the crack of the door in such a way that normal traffic could be maintained. Where the doors were very closely fitted and this was not possible, it was necessary to block off one set of doors and crack them slightly to pass the tubing.

---

* The terms are defined in the Test Results.

Pressure differentials in the Cleveland buildings were recorded for 30-min intervals; and in some buildings, several recordings were made at different times during the day. In the Pittsburgh buildings, draft gage readings were taken at 15-sec intervals for 15 min.

Wind velocities were taken by a vane-type anemometer at the curb line in front of the entrance. Observations were made with the instrument first facing either windward, and then in a direction perpendicular to the entrance. The outside and inside wet-and dry-bulb temperatures were taken at the street level, the top of the building, and at 2 intermediate points.

Phase 2: For the second phase of the program, it was decided to investigate the flow coefficients of doors by a laboratory study of scale models. The equipment

FIG. 1—LABORATORY EQUIPMENT FOR STUDYING AIR FLOW THROUGH SCALE MODEL ENTRANCES

shown in Fig. 1 was constructed for the purpose. Models of single-bank doors were located in the face of the 4- x 5-ft test section, and an extension, shown in place in the figure, provided the location for a second set of doors for studying vestibule-type entrances. Air which entered through the doors passed through a bellmouth transition and a metering section to the fan inlet.

The static pressure in the test section was controlled by adjusting a cone-shaped damper in the fan discharge. The pressure inside the entrance was sensed through small holes in the top of a copper tube placed horizontally about 24 in. back of the doors and 9 in. above the floor.

The magnitude of the pressure was determined by the pressure transducer and recorder. Air flow was measured by an orifice plate in the metering duct. The entire setup was tested for leakage and proved tight.

To determine the feasibility of scale-model tests, a preliminary investigation was made with 1/6 and 1/4 scale models of a 3- x 7-ft single-door entrance. The results obtained with these models were found to agree well with results obtained from tests of a full scale door in the Laboratory.

In addition to the 2 single-door models just mentioned, a 1/6 scale model of a 4-door entrance was constructed and tested. In this model, a pair of 2½- x 7-ft, opposite-swinging doors was located at the center, and at each end was a 3- x 7-ft

side-swinging door. Two identical panels were made of each arrangement, so that each could be tested as a single bank or vestibule type of entrance. In each entrance model, not only the doors, but the jambs, thresholds, and other details were constructed to scale.

All tests were made under steady conditions with doors set at various angular positions. For each combination of door positions, air flow was determined for at least 3 pressure differentials across the model entrance. Wet- and dry-bulb temperatures were measured by thermometers located in the metering duct.

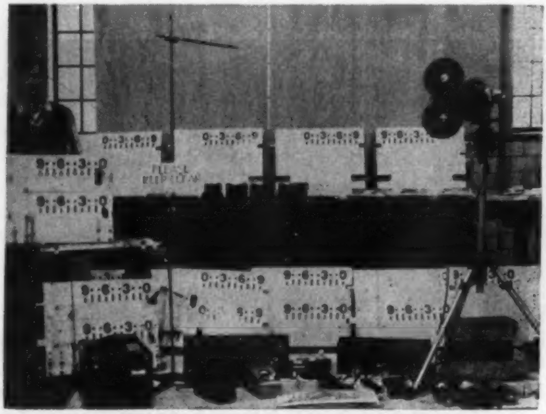


FIG. 2—EQUIPMENT USED IN FIELD TESTS OF BUILDING ENTRANCES

Phase 3: Studies on the correlation of door position and traffic rate were conducted in 7 buildings in the downtown area of Cleveland during November and December 1956, and January 1957. One of the entrances studied had power-operated doors, one set of which swung in, and the other set of which swung out. The other 6 buildings had manually operated doors which swung out.

Indicators, shown in Fig. 2 with other field test equipment, were built to show the position of the entrance doors to be studied. Fig. 3 shows the indicators for both the inner and outer banks of vestibule doors, mounted over the inner doors. Indicators were connected to the doors by a system of cords and pulleys, and were calibrated in place before the start of a test.

A 16 mm movie camera operating at a speed of 8 frames per second was used to take pictures similar to Fig. 3, showing the indicators and the people passing through the doors. At this operating speed, the 400-ft films used would last for 30 min continuous operation. The camera was actually operated intermittently for periods of 4 or 5 min each, at various times throughout the day, so that data could be obtained for various traffic patterns.

After the films were developed, the pictures were analyzed with the aid of a time-motion-study projector. Pictures could be advanced, one frame at a time, and a counter indicated the number of the frame being studied.

TEST RESULTS

Pressure Differential Across an Entrance: It was found from an analysis of the field test data that the pressure differential across a building entrance was related to the building height, and the difference between the indoor and outdoor air temperatures. It was also evident that the differential could be increased or decreased appreciably by the operation of exhaust or fresh air supply fans.

To correlate the pressure differential across an entrance with building height, temperature differential, and blower action, the observed pressure differential for

FIG. 3—ENTRANCE SHOWING DOOR INDICATORS AND TRAFFIC

each test was expressed as a percentage of the theoretical natural draft for the conditions of the test. This ratio is hereinafter referred to as a ***draft-factor***.

The theoretical draft which is produced solely by the stack height and the difference in the densities of the cold outside and the warm inside air may be calculated by Equation 1.

$$\Delta P_{\text{theo.}} = 0.1924\ (\rho_o - \rho_i)\ H \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad (1)$$

The theoretical draft in buildings, heated to 75 F, of any height, and for any inside-outside temperature difference, may be determined from Fig. 4. The figure is based on air densities taken from Reference 5.

In Fig. 5, the observed pressure differentials resulting from chimney action only have been plotted against the theoretical drafts obtained from Fig. 4. A straight line drawn through the points for conventional buildings has a slope of 0.7, thus indicating a *natural draft factor*, f_N, of 0.7. One of the buildings tested was of metal curtain wall construction and had windows which were tightly sealed by inflated gaskets. This building had a draft factor of 0.3 as shown by the lower curve

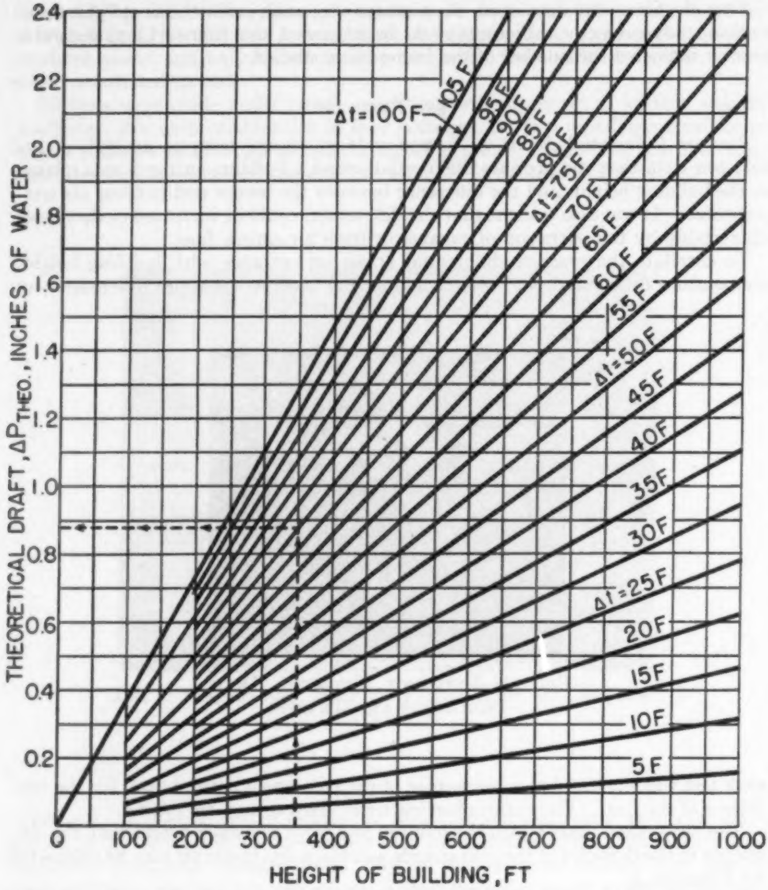


Fig. 4—Theoretical Draft in Tall Buildings Due to Chimney Effect (Based on Average Inside Air at 75 F, 50 Percent RH, Standard Atmospheric Pressure 29.921 in. Hg)

of Fig. 5. However, since only one building of this type was tested, it is not known if this lower factor would be typical for other buildings of similar construction.

The operation of exhaust or fresh air supply fans tends to increase or decrease the pressure differential across a building entrance caused by natural draft. Corrections for the natural draft factor, for various rates of air exhaust or supply, are given in Fig. 6. The corrected factors, or the *forced draft factors*, are also given in the figure.

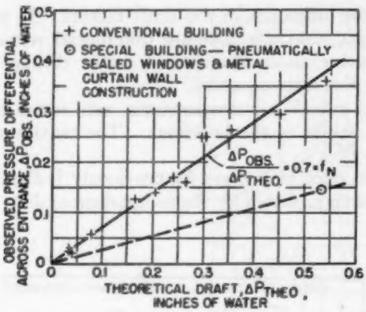


FIG. 5—NATURAL DRAFT FACTOR FOR BUILDING ENTRANCES

Wind velocity was found to have a negligible effect on the pressure differential in the congested business areas where the field tests were conducted. Although some tests were made on rather windy days when wind velocities of 20 mph were being recorded by the U. S. Weather Bureau, wind velocities measured windwardly at the curb line in front of the entrance were not more than 6 mph, and the maximum velocity normal to the entrance was only about 2 mph, which is equivalent to a pressure differential of less than 0.002 in. of water. The wind direction past the building entrance was found to be quite erratic.

No attempt was made to investigate the effect of vertical openings. However, the presence of large escalator openings in two of the buildings tested did not seem

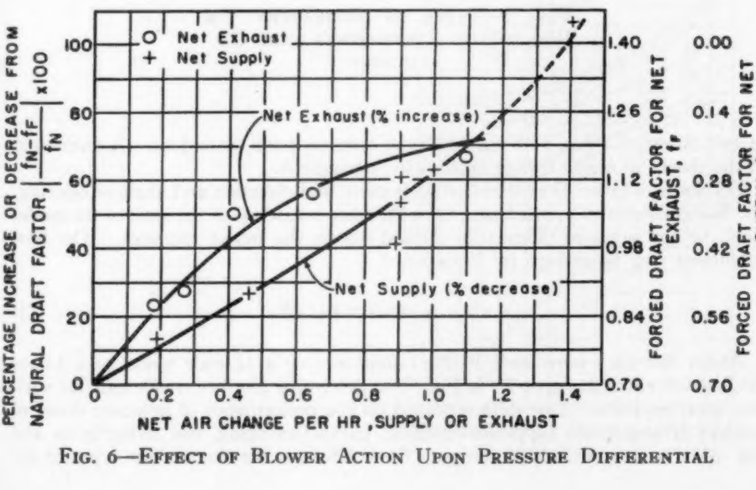


FIG. 6—EFFECT OF BLOWER ACTION UPON PRESSURE DIFFERENTIAL

to cause any significant difference in the draft factors. It was also surprising to the investigators to find that the pressure differential remained essentially constant regardless of variations in the rate of traffic through the entrances. No effect of *elevator pumping* was observed.

Both the inside and outside air temperatures were found to be reasonably constant over the entire height of the buildings. The maximum variation observed was of the order of 5 F.

Flow Coefficients for Door Openings: As previously indicated, the feasibility of scale model tests was demonstrated by the comparison of data obtained with models

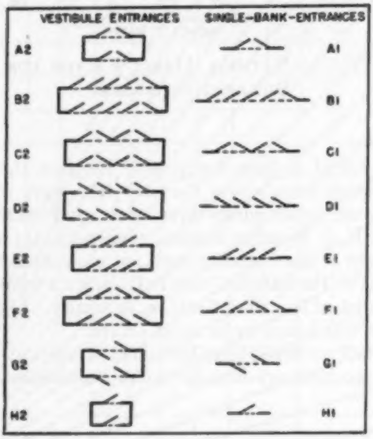


FIG. 7—TYPES OF ENTRANCES FOR WHICH FLOW COEFFICIENTS WERE CALCULATED

at two different scales, with the results of tests on a full scale door. A discussion of the theory of model testing appears in Appendix A.

For doors of given dimensions, arrangement, and direction and angle of opening, the flow coefficient, C_F, was found to be the same regardless of the scale of the model used, or the pressure differential applied across the model entrance. The flow coefficient may be defined by Equation 2.

$$C_F = q_n/4005\ NA\sqrt{\Delta P} \qquad (2)$$

About 300 tests were made in the Laboratory on a ⅙ scale model of a 4-door single-bank entrance (type F1 in Fig. 7) to determine the flow coefficients for various door positions. Test data obtained on the performance of adjacent doors of various arrangements (opposite-swinging, parallel-swinging, one swinging in and the other out) were used to compute the coefficients for the various types of en-

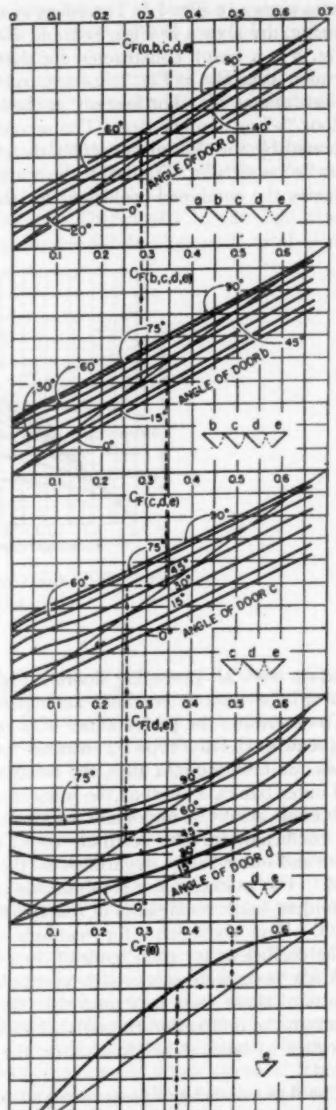


FIG. 8—CHART FOR DETERMINING FLOW COEFFICIENTS FOR TYPE B1 ENTRANCE

trances shown in Fig. 7. The effect of adjacent doors, either parallel- or opposite-swinging, are shown in Figs. B-1 and B-2 of Appendix B.

Fig. 8 is a graph constructed for determining the flow coefficients for a 5-door entrance (type B-1 of Fig. 7) for any combination of door positions. It is composed of 5 sets of curves. The first set, at the bottom of the figure, shows the performance of door "e" as a single door. The second set of curves, when entered at the point indicated by the "e" curves, gives the coefficient for the combined performance of doors "e" and "d". Similarly, the scale at the top of each successive set of curves indicates the combined coefficient for the door represented by the curves, in com-

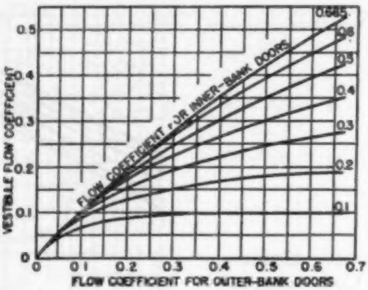


FIG. 9—FLOW COEFFICIENTS FOR VESTIBULE ENTRANCES

bination with all preceding doors. Fig. 8 is typical of a number of graphs constructed for various entrance arrangements.

To determine the performance of a vestibule-type entrance, tests were made on a ⅙ scale model of a type F2 entrance having a vestibule depth of 8 ft in prototype. It was at first thought that the relative locations of the open doors in the inner and outer banks, as well as the numerous combinations of positions of the doors in each bank, might be important variables in the study. However, it was found that the coefficient for the vestibule could be predicted from the coefficients for the inner and outer banks, regardless of the positions of the individual doors in each bank. For example, any combination of door positions in the 2 banks which gave individual bank coefficients of 0.3 and 0.4 would produce a vestibule coefficient of 0.225. Flow coefficients for vestibule entrances may be determined from Fig. 9, if the coefficient for each bank of doors is known.

A few tests were made with vestibules 4 ft and 16 ft deep in prototype. The results of these tests indicated that the depth of the vestibule had no significant effect on the infiltration through a building entrance. Tests were also made with a marquee at several heights above the entrance, but no significant effects were found.

The flow coefficients as determined from Figs 8 and 9 are for doors at the indicated positions but unobstructed by people passing through them. The effect of human obstruction was studied by placing a scaled model of human figure at various positions in a door opening. The results of these tests are given in Appendix

C. Other tests, in which 3 human figures were placed at various positions in an open doorway indicated that the entire effect observed was created by the one figure which created the maximum obstruction.

Since the reduction of infiltration due to human obstruction depends upon the length of time the opening is obstructed, further discussion on the subject will be given with the results of the traffic analysis.

A few tests were made to determine the infiltration through a full scale door crack 34½ in. long and adjustable in width up to ½ in. The results are given in Fig. 10.

Analysis of Traffic and Evaluation of Entrance Coefficients: In the field studies made to correlate traffic and door position, seven 400-ft films or a total of over

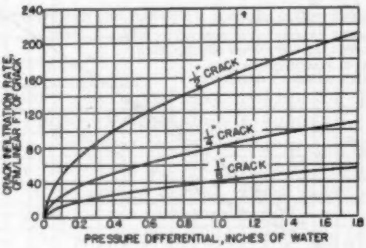


FIG. 10—INFILTRATION THROUGH DOOR CRACKS

100,000 frames were taken at a speed of 8 frames per second. These pictures showed traffic at various rates passing through vestibule-type entrances having various door arrangements and vestibule depths. It was found that good accuracy could be obtained by the analysis of every eighth frame (1 picture per second), and the results presented are therefore based on the examination of approximately 12,500 frames.

The analysis was made by projecting every eighth frame onto a screen, and noting the indicated position of each door in both the inner and outer banks. For each combination of door positions, the flow coefficient for each bank of doors was determined from a chart similar to Fig. 8, constructed for the particular entrance being studied. The traffic rates through the doors for certain selected sequences were also determined from the pictures.

From the data obtained from the pictures, *entrance coefficients* were calculated for the various entrances tested. An entrance coefficient may be defined as the flow coefficient corresponding to a given traffic rate through a given entrance.

Entrance coefficients for a single-bank entrance were determined by averaging the flow coefficients for the outer bank of vestibule entrance doors. A plot of these entrance coefficients *vs.* traffic rate is shown by the upper curve of Fig. 11. The flow coefficients for the outer bank of doors were used because it was felt that the outer bank more nearly represented the performance of single bank doors. In every building tested, the outer doors closed more rapidly than the inner doors, probably due in part to the adjustment of the door closers, and in part to the

greater pressure differential acting on the outer doors. Field tests had indicated that most of the pressure differential across a vestibule-type entrance occurred at the outer set of doors. Coefficients based on the inner doors would have been about 10 percent higher than those shown in Fig. 11.

Flow coefficients for vestibule-type entrances were determined from the separate flow coefficients for the inner and outer banks of doors by means of Fig. 9. The vestibule entrance coefficients shown by the upper curve of Fig. 12 were determined by averaging the vestibule flow coefficients for given traffic rates. An example of the determination of entrance coefficients is given in Appendix D.

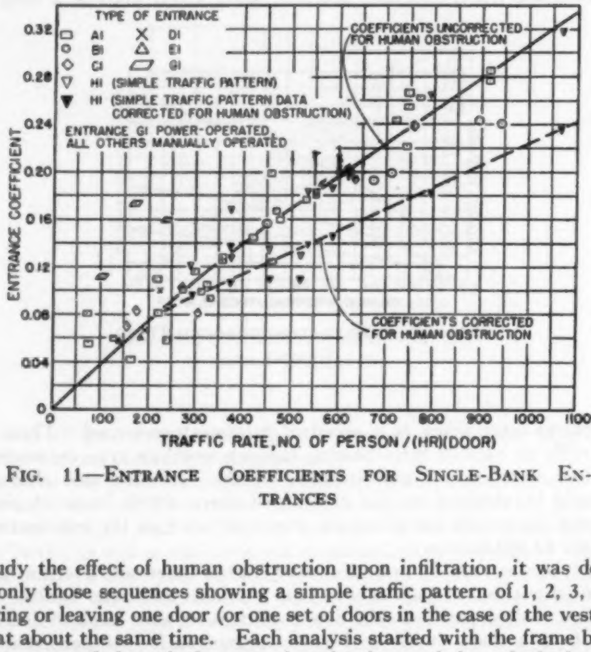


FIG. 11—ENTRANCE COEFFICIENTS FOR SINGLE-BANK ENTRANCES

To study the effect of human obstruction upon infiltration, it was decided to analyze only those sequences showing a simple traffic pattern of 1, 2, 3, or 4 people entering or leaving one door (or one set of doors in the case of the vestibule entrance) at about the same time. Each analysis started with the frame before the first person entered through the outer door (or departed through the inner door), and continued until the last person had completed the passage through the doors, and the doors were closed. The angles of door opening, as well as the position of each person relative to the doors, were noted; and the effects of the obstruction were determined and can be seen from Fig. C-1 of Appendix C. The lower curves of Figs. 11 and 12 were plotted on the assumption that the ratio of the corrected to the uncorrected entrance coefficients, as found for the simple traffic patterns studied, would hold for the more complex patterns. As shown in the figures, the effect of human obstruction is negligible at low traffic rates, and increases to approximately 25 percent with 1,000 persons per hour passing through each door.

The solid curves in Figs. 11 and 12 were drawn to represent the average coefficients for all of the entrances tested. A close inspection of the figures reveals that the

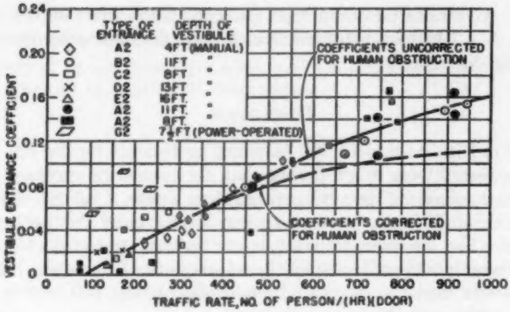


FIG. 12—ENTRANCE COEFFICIENTS FOR VESTIBULE ENTRANCES

solid curves also represent a reasonably good average for each of the entrances tested except those having power-operated doors, G1 and G2. It is therefore recommended that the entrance coefficient for any entrance having manually operated doors be determined from the curves of Figs. 11 and 12, regardless of door size or arrangement. The use of the broken curves, giving values corrected for human obstruction, is recommended for practical application.

The curves of Fig. 13 show the relationship between the effective (actual) pressure differential across an entrance, the entrance coefficient, and the infiltration

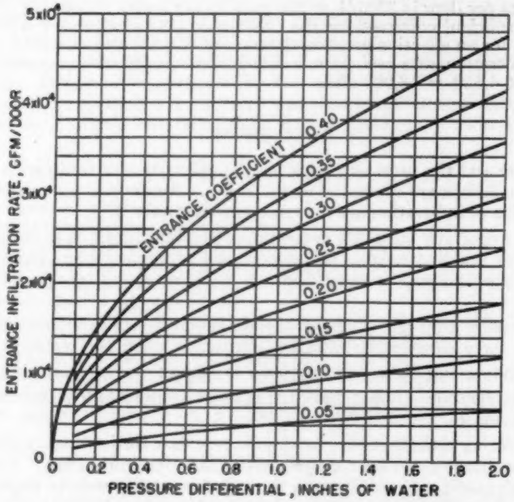


FIG. 13—ENTRANCE INFILTRATION RATES FOR VARIOUS PRESSURE DIFFERENTIALS AND TRAFFIC RATES

rate per door. It should be noted that the infiltration rates shown in this figure are based on a 3- x 7-ft door. Infiltration through doors of other sizes will be proportioned to the door areas.

Discussion

As indicated in Figs. 11 and 12, the entrance coefficients for the one set of power-operated doors tested were much higher than for manually operated doors. This is to be expected, since the power-operated door opens a full 90 deg each time a person passes through it, and remains open longer than the average manually-operated door. The increase in infiltration through power-operated doors will probably vary widely, depending upon the adjustment of the operating mechanism. It is also apparent that the effect of human obstruction will be less with power-operated than with manually-operated doors.

The data on infiltration through door cracks will be applicable only to entrances through which the traffic rate is very low.

Illustrative Example: *Given*: *A conventional building 350 ft high has a net cubage of 4,200,000 cu ft. Fresh air fans supply outside air at the rate of 80,000 cfm, and exhaust fans discharge 50,000 cfm. The building has a vestibule-type entrance having four 3- x 7-ft doors and the inside temperature is maintained at 75 F. Find*: *Infiltration rate through the entrance at a time when the outside temperature is zero F, and the traffic rate is 2,000 persons per hour.*

Solution: From Fig. 4, the theoretical draft for a 350-ft building height and a 75 F inside-outside temperature differential is 0.88 in. of water.

The net air supply rate is 80,000 − 50,000 = 30,000 cfm. This is 30,000 × 60/4,200,000 or 0.43 air changes per hour. From Fig. 6, the draft factor for a conventional building, corrected for an air supply rate of 0.43 air changes per hour is 0.525, and the effective pressure differential is 0.88 × 0.525 = 0.462 in. of water.

The traffic rate per door is 2000/4 = 500 persons per hour. From Fig. 12, the vestibule entrance coefficient corrected for the effect of human obstruction is 0.08.

From Fig. 13, for an effective pressure differential of 0.46, and an entrance coefficient of 0.08, the infiltration rate per door is 4,500 cfm. Infiltration through the 4-door entrance = 4 × 4,500 = 18,000 cfm.

Conclusions

1. A procedure and the necessary data for the calculation of infiltration through entrances of tall buildings are presented in the paper. An approximate general equation for the calculation of infiltration are also given in Appendix E.

2. The most important variables determining the infiltration rate through building entrances are (*a*) building height, (*b*) indoor-outdoor temperature difference, (*c*) the quantity of air supplied and/or exhausted, (*d*) traffic rate, and (*e*) type of building entrance.

3. The infiltration through a vestibule-type entrance varies from about 50 to 60 percent of that for a single-bank entrance, depending upon the traffic rate.

4. Wind velocity is not an important factor in determining entrance infiltration.

5. Entrance infiltration is not appreciably affected by the depth of the vestibule or the presence of a marquee.

6. It is only during periods of very low traffic rate that the infiltration through cracks around the doors becomes an appreciable part of the total infiltration. During periods of normal or high traffic rates, infiltration through door cracks may be neglected without serious error.

7. The approximate entrance infiltration in a 30-story building at a temperature differential of 75 F is as follows:

For single-bank door entrances—900 cu ft per person per passage.

For vestibule-type entrances—550 cu ft per person per passage.

A net air supply of ½ air change per hour, and a traffic rate of 500 persons per (hour) (door), were assumed in arriving at these values.

ACKNOWLEDGMENTS

The author wishes to express his sincere thanks to the following: To Professors Alfred Koestel and G. L. Tuve of Case Institute of Technology for their helpful suggestions. To The Sandborn Co., International Steel Co., and Decker Aviation Corp. for lending instruments which greatly facilitated the collection of data. To the owners, managers, and engineers of the following *buildings*, for their helpful cooperation in the field-testing phases of the program, located in:

Pittsburgh: Grant, Koppers, U. S. Steel, Oliver, Alcoa, Gateway Center, Gulf and the Cathedral of Learning at University of Pittsburgh.

Cleveland: Fenn Tower at Fenn College, Leader, Midland, Republic, Guildhall, Standard, Schofield, Terminal Tower, The May Co., Cuyahoga Savings, Standard Drug Co., Sterling Lindner Davis, The Capital Bank, The Bailey Co., and Halle Brothers.

REFERENCES

1. The Infiltration Problem of Multiple Entrances, by A. M. Simpson and K. B. Atkinson (ASHVE JOURNAL SECTION, *Heating Piping & Air Conditioning*, Vol. 8, No. 6, June, 1936, pp. 345–351).

2. Infiltration Characteristics of Entrance Doors, by A. M. Simpson (*Refrigeration Engineering*, Vol. 31, No. 6, June, 1936, p. 345).

3. Heat Transmission and Infiltration through Doors, Windows and Glass Masonry, American Architect Time-Saver Standards Series, No. 66. (*American Architect and Architecture*, Vol. 149, No. 2651, Nov. 1936, p. 95).

4. New Data on Entrance Door Infiltration with Air Conditioning, by A. M. Simpson (*American Architect and Architecture*, Vol. 149, No. 2651, Nov. 1936, pp. 85–88).

5. HEATING VENTILATING AIR CONDITIONING GUIDE, 1956 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, pp. 32–39).

NOMENCLATURE

A = area of a door, square feet.
C_F = flow coefficient, dimensionless.
f_F = forced draft factor, dimensionless.
f_N = natural draft factor, dimensionless.
H = building height, feet.
N = number of doors per bank.
ΔP = pressure differential across the entrance, inches of water.
$\Delta P_{obs.}$ = observed pressure differential, inches of water.
$\Delta P_{theo.}$ = theoretical pressure differential, inches of water.
q_n = air flow through entrance, cubic feet per minute.
ρ_i = density of inside air, pounds per cubic foot.
ρ_o = density of outside air, pounds per cubic foot.

APPENDIX A

PRELIMINARY INVESTIGATION ON MODEL ENTRANCE TESTS

Before the model tests could be of any value, investigation was made to determine how a geometrical, kinematic, and dynamic similarity could be established between

the prototype and the model, so that the phenomena of the prototype could be reproduced in the model, and the information obtained from the model could be used to predict the performance of the prototype.

From the knowledge of the physical situation, it was assumed that the controlling factors for the air volume flow through the door entrance would be ΔP, D, ρ, g, and μ, and that the air volume flow q might be expressed by the equation:

$$q = \phi(\Delta P, D, \rho, g, g_c, \mu) \qquad \text{(A-1)}$$

where

q = air infiltration, cubic feet per minute
ϕ = any function
ΔP = pressure differential, pounds$_f$ per sq ft
D = characteristic dimension, feet
ρ = air density, pounds$_f$ per cubic foot
g = acceleration due to gravity, feet per minute (minute)
g_c = conversion factor, $\dfrac{(\text{lb}_m)(\text{ft})}{(\text{lb}_f)(\text{min})(\text{min})}$
μ = air viscosity, $\dfrac{\text{lb}_m}{(\text{ft})(\text{min})}$

By the use of the dimensional analysis, Equation A-1 becomes:

$$\frac{q/D^2}{\sqrt{2g_c\dfrac{\Delta P}{\rho}}} = a_1\left(\frac{D\rho\sqrt{2g_c\dfrac{\Delta P}{\rho}}}{\mu}\right)^{a_2}\left(\frac{\Delta P g_c}{D\rho g}\right)^{a_3} \qquad \text{(A-2)}$$

where

a_1 = experimental constant
a_2 = experimental exponent
a_3 = experimental exponent

Since D^2 may be replaced by any convenient dimension of area such as the area of a door A, Equation A-2 may be written as

$$\frac{q}{A\sqrt{2g_c\dfrac{\Delta P}{\rho}}} = C_F = a_1\left(\frac{D\rho\sqrt{2g_c\dfrac{\Delta P}{\rho}}}{\mu}\right)^{a_2}\left(\frac{\Delta P g_c}{D\rho g}\right)^{a_3} \qquad \text{(A-3)}$$

where

$$C_F = \text{flow coefficient}$$

or

$$N_{Eu} = a_1(N_{Re})^{a_2}(N_{Fr})^{a_3} \qquad \text{(A-4)}$$

where

$$N_{Eu} = \frac{q}{A\sqrt{2g_c\dfrac{\Delta P}{\rho}}} = \text{Euler Number}$$

$$N_{Re} = \frac{D\rho}{\mu}\sqrt{2g_c\frac{\Delta P}{\rho}} = \text{Reynolds Number*}$$

$$N_{Fr} = \frac{\Delta P g_c}{D\rho g} = \text{Froude Number*}$$

* *Fluid Mechanics for Hydraulic Engineers*, by Hunter Rouse (McGraw-Hill Book Company, Inc., New York, First Edition, 1938, p. 303).

As dimensional analysis can serve only as a yardstick to reveal the possible parametric groups which may govern the characteristics of the flow, the final answer must rely on the test results, first to determine the effect of the Reynolds and the Froude Numbers on the Euler Number, and then to determine the experimental constants a_1, a_2, and a_3.

If experiments showed that both the Reynolds and Froude Numbers were important, the flow coefficient would have to be split into two parts†, one to be a function of the Reynolds Number, and the other the Froude Number.

Experiments were run in the wind tunnel with a ¼ scale single-door entrance as prototype and ⅙ scale single-door entrance as the ⅔ scale model, both at the fully open position. By proper choice of the size of doors and selection of the pressure differentials, either the Reynolds Number or the Froude Number was held constant, and the Euler Number, in either case, was found to be the same. This pointed out that both the

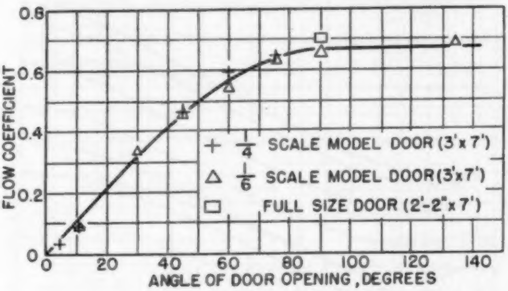


FIG. A-1—FLOW COEFFICIENTS FOR A 3- × 7-FT DOOR AT VARIOUS ANGLES OF OPENING

Reynolds and Froude Numbers were unimportant, and that the Euler Number, or the flow coefficient, depended only upon the geometry of the boundary, *i.e.*, for a given geometrical orientation, the average velocity was proportional to the square root of the pressure differential, and the proportionality constant was the same for both the prototype and model regardless of the pressure differential and the size of the scale model. In other words, the experimental exponents a_2 and a_3 in Equation A-4 were zero, and the value of a_1 remained the same for a given boundary orientation.

The fact that the Euler Number remained constant for a given boundary condition indicated that the Reynolds and Froude Numbers were sufficiently large ($N_{Re} > 10^5$, $N_{Fr} > 2$) that the viscous and gravity forces were negligible in comparison to the inertia and pressure forces*.

Existing data** show that the Euler Number or flow coefficient for a circular orifice is constant at a Reynolds Number of 1×10^5 and higher. The literature also shows that, for a flow with free surface, the Euler Number is constant for a Froude Number of 2 and greater. In the case where the flow is not free but guided by a horizontal plane, such as in a door entrance with floor simulation, the gravity force is not important, and

† *Fluid Mechanics*, by Russell A. Dodge and Milton J. Thompson (McGraw-Hill Book Company, Inc., New York, First Edition, 1937, p. 436).

* *Fluid Mechanics for Hydraulic Engineers*, by Hunter Rouse (McGraw-Hill Book Company, Inc., New York, First Edition, 1938, pp. 303, 304).

** *Elementary Mechanics of Fluid* by Hunter Rouse (John Wiley & Sons, Inc., New York, 1946, pp. 168, 105, 93, 95).

the flow coefficient is independent of the Froude Number and depends only upon the boundary proportions and orientations. This is also found to be true for a sluice gate††.

Since the wind tunnel tests were run at room temperature, the properties of air were essentially constant. Only the pressure differential ΔP and scale of the model D were varied. For a single-door entrance of ⅙ scale at a pressure differential even as low as 0.02 in. water, the corresponding Reynolds and Froude Numbers were 4×10^3 and 2 respectively. The pressure differentials used in all 700 tests ranged from 0.05 to 0.98 with most tests made around 0.5 in. water, which was the practical range encountered in the field tests. The Reynolds and Froude Numbers were therefore greater than their critical values in all tests.

The flow coefficient for both ¼ scale and ⅙ scale single-door entrances was found to be about 0.665 for the fully open position. A check test was made in a corridor by

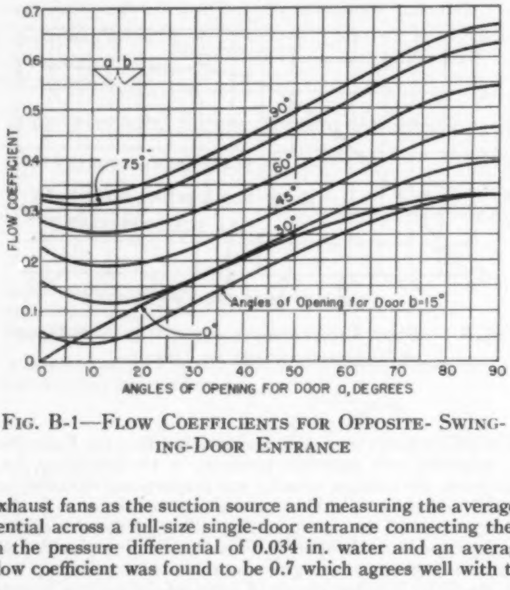


FIG. B-1—FLOW COEFFICIENTS FOR OPPOSITE- SWINGING-DOOR ENTRANCE

using 2 large exhaust fans as the suction source and measuring the average velocity and pressure differential across a full-size single-door entrance connecting the corridor and a room. With the pressure differential of 0.034 in. water and an average velocity of 520 fpm, the flow coefficient was found to be 0.7 which agrees well with the model test value of 0.665.

Tests were run on both ¼ and ⅙ scale models at different angles of opening, and the results showed that the flow coefficients[1] remained the same for both scales at the same angles of opening. This is shown in Fig. A-1.

APPENDIX B

EFFECT OF THE DIRECTION OF SWING OF ADJACENT DOORS ON THE FLOW COEFFICIENT

The effect of the direction of swing of adjacent doors was investigated so that the flow coefficient for various types of entrances (Fig. 7) could be evaluated. Fig. B-1

†† *Elementary Fluid Mechanics* by John K. Vennard (John Wiley & Sons, Inc., New York, Third Edition, 1954, p. 305).

[1] The flow coefficients for a door at various angles of opening were based on the full area of the door, as defined in Equation 2, instead of the actual area of the opening of the door.

shows the flow coefficients for a pair of opposite swinging doors, and Fig. B-2 shows the coefficients for a single door as influenced by an adjacent parallel-swinging door.

Tests indicated that, for an opposite-swinging-door entrance with one door swinging out and the other in, the flow coefficient was essentially the same as for a parallel-swinging-door entrance with both doors swinging out.

APPENDIX C

EFFECT OF HUMAN OBSTRUCTION

To evaluate the reduction of air infiltration through an entrance due to the obstruction of people passing through, tests were run in the wind tunnel with scaled figures of

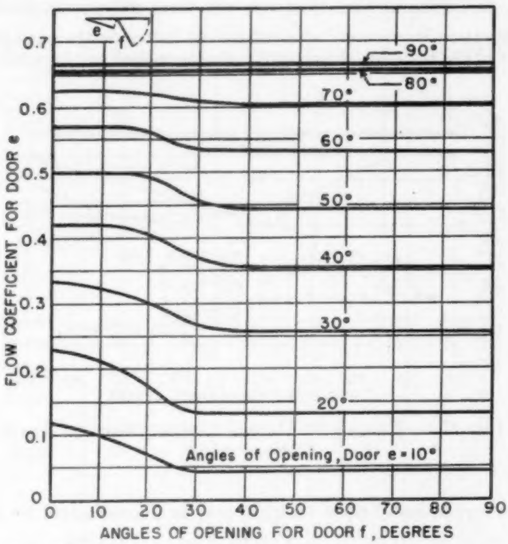


FIG. B-2—FLOW COEFFICIENTS FOR DOOR e SHOWING INFLUENCE OF ADJACENT DOOR f

persons standing at the model single-bank-door entrance. The results are shown in Fig. C-1.

For vestibule entrances, corrections for human obstruction may be computed by the use of Figs. 8, C-1, and Fig. 9. For instance, at a single-door vestibule entrance with one subject standing at the threshold of the outer door and the other at the threshold of the inner door, and both inner and outer doors at fully open position, the flow coefficients for inner door and outer door with correction for human obstruction are $0.665 \times 0.67 = 0.45$ (Figs. 8 and C-1 for position 4 curve at 90 degrees). The vestibule coefficient is 0.31 (from Fig. 9) which agrees well with the test value of 0.33.

APPENDIX D

SAMPLE CALCULATION OF ENTRANCE COEFFICIENT

The entrance coefficient for each sequence of frames analyzed was determined by averaging the flow coefficients for the sequence.

Table D-1 is a sample of the data developed by analyzing a sequence of pictures. By dividing the totals for the sequence by the number of frames in the sequence, 266, the following results are obtained.

$$\text{Single-bank entrance coefficient} = \frac{51.704}{266} = 0.194$$

$$\text{Vestibule-entrance coefficient} = \frac{28.412}{266} = 0.107$$

APPENDIX E

APPROXIMATE GENERAL EQUATION FOR THE CALCULATION OF INFILTRATION

Entrance infiltration is primarily affected by the indoor-outdoor temperature difference, the height of the building, the quantity of air supplied and/or exhausted, and the

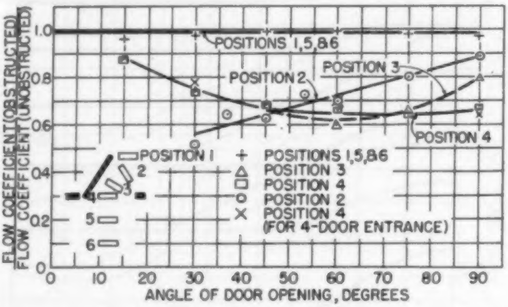


FIG. C-1—EFFECT OF HUMAN OBSTRUCTION ON FLOW COEFFICIENTS

traffic rate. A correlation of these variables may be approximated by the equation:

$$q' = k_1(M + k_2)\left[\frac{H}{T_o}(\Delta t)(1 + k_3 N^{k_4})\right]^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad \text{(E-1)}$$

where

q' = entrance infiltration corrected for human obstruction, based on inside design temperature of 75 F, cubic feet per minute per door
M = traffic rate, number of persons per (hr)(door)(limited to 200 to 1,000)
H = building height, feet
T_o = Absolute outside air temperature, Rankine degrees
N = net air changes per hour, (exhaust or supply) (limited to 0 to 1.1)
k_1 = constant, 1.1 for vestibule entrance; 1.6 for single-bank-door entrance
k_2 = constant, 125 for vestibule entrance; 216 for single-bank-door entrance
k_3 = constant, −0.6 for net supply air; +0.7 for net exhaust air
k_4 = exponent, 0.9 for net supply air; 0.6 for net exhaust air
Δt = indoor-outdoor temperature difference, Fahrenheit degrees

DISCUSSION

H. T. GILKEY, Cleveland, Ohio (WRITTEN): Having been associated with the Technical Advisory Committee on Heating and Air Conditioning Loads during the period of

Table D-1—Analysis of a Type B Entrance With A Traffic Rate of 674 Persons per (Hr)(Door)

Frame No.	Bank	Angles of Doors[a]					Flow Coefficients	
		A	B	C	D	E	Single Bank	Vestibule
3137	outer	18	0	30	84	40	0.244	0.167
	inner	65	48	0	36	0	0.276	
3145	outer	8	0	20	65	14	0.150	0.131
	inner	68	64	0	30	0	0.295	
3153	outer	2	0	10	16	5	0.047	0.047
	inner	66	59	0	30	20	0.276	
3161	outer	4	0	0	10	0	0.044	0.044
	inner	44	60	0	22	61	0.292	
3169	outer	4	0	0	20	0	0.054	0.054
	inner	17	58	0	22	90	0.279	
3177	outer	4	0	0	48	36	0.116	0.122
	inner	9	32	0	30	76	0.210	
⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮
4609	outer	38	50	0	8	0	0.179	0.137
	inner	50	0	4	40	74	0.259	
4617	outer	25	60	0	4	0	0.174	0.133
	inner	64	0	0	20	76	0.249	
4625	outer	54	54	0	0	0	0.210	0.142
	inner	56	0	0	4	74	0.232	
⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮
5241	outer	2	0	0	0	26	0.060	0.060
	inner	45	15	0	38	0	0.194	
5249	outer	2	0	0	0	10	0.034	0.033
	inner	20	10	0	38	0	0.133	
5257	outer	2	0	0	0	4	0.015	0.015
	inner	10	5	0	38	0	0.107	
5265	outer	2	0	0	0	4	0.015	0.015
	inner	6	0	0	40	0	0.095	
	Total of coefficients for outer bank						51.704	
	Total of coefficients for vestibule entrance							28.412

[a] See Fig. 8 for door designations.

the investigation reported in this paper, perhaps I am in a better position than some to express my appreciation to Mr. Min and the staff of the Research Laboratory for the information presented. Quite frankly, several members of the TAC were initially skeptical that information of practical value could come from such an investigation. I think that the committee was somewhat overwhelmed by the number of factors which might affect entrance infiltration. Many, but not all of those, are listed in the paper. It is true that some of these variables were found to be of negligible importance during the course of the investigation, but this fact detracts in no way from the skill and ingenuity exhibited in conducting the investigation, and in analyzing and interpreting the data.

It should be pointed out that a great many types of structures are grouped within the classification of *conventional buildings* shown in Fig. 5. This classification includes buildings of curtain-wall construction, as well as of masonry construction. The paper specifically mentions that one of the buildings tested was of metal curtain-wall construction and had windows which were tightly sealed by inflated gaskets; this building had a much lower draft factor than did the other buildings investigated. Obviously, not all metal curtain-wall buildings are as tightly sealed against air leakage as was this one, and the lower natural draft factor was probably due to the tightly sealed windows rather than to the curtain-wall construction as such.

The report states that the pressure differential across the building entrance remained essentially constant regardless of variations in the rate of traffic through the entrances. One wonders if this situation was observed only during the day, when there would always be at least light traffic, or was it also observed in the evening at an essentially zero-traffic condition?

Obviously, the entryway is not a building's only source of air leakage. There is some question, I think, as to whether the entryway represents the major source of infiltration in buildings such as those studied, or if it is merely a source of substantial air leakage. Unfortunately, this question is not resolved by the observation *that the pressure differential remained essentially constant regardless of variations in the rate of traffic through the entrances*. It is wondered, therefore, for the building sited in the illustrative example, if there are means of estimating the proportion of total air leakage which the 18,000 cfm entrance leakage represents.

It is unfortunate that the appendices could not be published with the paper in the JOURNAL, for without them complete analysis of the paper is impossible. Furthermore, these appendices explain techniques of investigation and analysis which will prove valuable to other investigators.

R. A. PARSONS, Lansing, Mich., (WRITTEN): The information provided by this paper should be helpful in determining infiltration rates through swinging type doors at street level entrances to large buildings. It is applicable to large buildings in congested areas during business hours. This data is necessary for computing the capacity of equipment needed to properly temper outside air as it enters such buildings.

Additional information would be useful if it were applicable to isolated buildings which do not have the benefit of surrounding buildings to minimize the effect of wind. Included might be such items as:

(1) Effectiveness of an *air curtain* in reducing infiltration. It has been stated that this type of installation seals entrances hermetically even against heavy winds, without producing a draft.

(2) The effect of exhausting air at entrances (pressurizing). Minimum volumes and velocities of exhaust air necessary to eliminate infiltration at various wind velocities could be determined.

D. R. BAHNFLETH, Urbana, Illinois (WRITTEN): The author has presented useful infiltration data in a form which is easily understood and easily applied. Many of the variables that influence infiltration through entrances have been studied and their rela-

tive importance established. However, it does seem that the omission of resistance to air flow between floors and through the building envelope should be given some consideration as an important variable. The *draft-factor*, which is the ratio of observed pressure difference at entrance level to theoretical pressure difference for a stack of the same height as the building, is a function of the resistance to air flow between floors and the resistance to air flow through the building envelope. A high draft-factor, which is indicative of a high pressure differential at entrance level, implies low resistance and a low draft-factor high resistance. Thus, if a structure is extremely tight above the level of the entrance, the pressure differential and entrance infiltration would be small.

To illustrate the effect of the resistance to air flow on infiltration consider the extreme case of an air-tight structure of 500-ft height that has a swinging door entrance. At an indoor-outdoor temperature difference of 75 F the theoretical pressure difference from Fig. 4 would be 1.25 in. water. The *draft-factor* would be nearly zero, and infiltration through the entrance would be negligible. From the results it is apparent that in conventional construction today the flow resistance between floors and the flow resistance of the envelope is very small, and no significant differences in draft-factor could be measured. The draft-factor of 0.7 places the neutral zone for infiltration at about 0.7 of the height, which is in agreement with earlier studies of heat loss in multi-story buildings. However, as new construction materials and new construction methods are introduced, the conventional building may have a greater resistance to flow and lower entrance infiltration rates. This is suggested by the results from the one test conducted on a presumably tighter-than-average building.

It should also be noted that the reduction of entrance infiltration can only be accomplished by (1) selecting the proper type of entrance, (2) controlling the net amount of ventilation air supplied to the structure, and (3) reducing the pressure differential across the entrance by increasing the resistance to flow of the building envelope, as such things as traffic rate, height and temperature difference cannot be controlled. Thus, if minimum entrance infiltration is desirable some attention must be given to the tightness of other components of the structure.

For the reasons mentioned, it would seem that the resistance of the building interior and envelope to air flow should be considered as one of the important variables determining entrance infiltration, and it would seem advisable to extend the present investigation, if possible, to evaluate draft factors for newer types of structures.

C. M. HUMPHREYS, Cleveland, Ohio: In his discussion Mr. Gilkey stated that some of the members of the Technical Advisory Committee doubted if the many variables associated with the problem of entrance infiltration could be resolved. These misgivings were also shared by some at the Laboratory. Mr. Min is to be commended for the work which he did, and particularly for his perseverance and resourcefulness in the analysis of the data.

As Mr. Min has stated, tests were made on buildings in both Cleveland and Pittsburgh. It is pleasant to take this opportunity to thank the owners, managers and engineers of these various buildings. In every case they offered their complete cooperation. Without this cooperation the project could not have been completed.

A. G. WILSON, Ottawa, Canada: I would like to add my commendation to Mr. Min and his colleagues at the ASHAE Laboratory for an excellent piece of work and one which I am sure will be regarded as highly significant. Having looked at the infiltration chapter of THE GUIDE recently, I realize how little information there is of the type that is presented in this paper. This information will be directly useable I am sure by the consulting engineer.

As has already been implied by some of the discussers the paper suggests further work that might be carried out, as is often the case. One of the subjects that has been discussed is the matter of the pressure differences across the entrances and the extent to which they approach the theoretical draft of the building. This is one aspect of the work which I think might usefully be extended to include pressure differences across the

exterior walls of buildings at various heights. I believe this would provide much needed information on pressure differences leading to infiltration into spaces at other levels in the building.

Finally, I would like to ask Mr. Min, to what extent he thinks the entrance flow coefficients obtained might be applicable to the determination of infiltration through entrances in the summer time in air-conditioned buildings.

W. R. RATAI, Milwaukee, Wis.: Fig. 8 is a chart for determining flow coefficients for type B-1 entrance. The angles, I presume, were measured by test or observed from test or from frames. Is any more work going to be done on any type of entrance such as plotting the coefficient for various types of entrances so one can determine the entrance which gives the lowest coefficient? As a previous discusser brought out, there are actually only two ways of decreasing or combatting infiltration. One is changing the entrance and the other is by increasing the net supply of air.

The second question is this. In any of these buildings was there any attempt at building up a positive pressure in the building by static-pressure-regulating the exhaust and supply fans?

Other questions I have are in regard to elevators. What type of elevators were there? How many were there? Did they have sealed doors?

Another question is in regard to the type of building—whether of masonry construction, 8-in. blocks or 4-in. brick or plaster, or what was the type of building construction.

Also in the equation, an area is shown. Is that area based on the total area of the door opening or is it based on the angular area, when the door is opened at a certain percentage angle?

E. C. MILES†, Pittsburgh, Penna.: I want to ask why the tightness of the building is not indicated as one of the important factors affecting the amount of infiltration, since it would appear that if a building were tight enough there wouldn't be any infiltration—no matter how tall, or how much updraft there might be.

AUTHOR'S CLOSURE: In answering Mr. Gilkey and Mr. Wilson's questions, it must first be said that it is most fortunate that this project has such a fine group on the Technical Advisory Committee who are not only able and experienced but also active and helpful. Several of the members are pioneers in the field of infiltration. This work represents, indeed, a joint effort.

As pointed out by Mr. Gilkey, this paper does not include information as to the proportion of the infiltration from other parts of the building. There are many problems in air infiltration. The paper, as its title implies, deals only with winter infiltration through swinging-door entrances in multi-story buildings. The investigation of many other problems such as infiltration at various levels of the building, infiltration through revolving-door entrances, entrance exfiltration under summer cooling conditions, and infiltration through various modern types of windows, is either being planned or being considered by the Technical Advisory Committee on Heating and Air-Conditioning Loads.

Mr. Wilson asks to what extent the entrance coefficients obtained under winter heating conditions might be applicable to summer cooling conditions. In the *wind tunnel* with scale-model entrances, it was found that flow coefficients for swinging-door entrances under summer cooling conditions are essentially the same as those under winter heating conditions. This was mentioned in the progress report to the Director of Research, ASHAE Research Laboratory, for the month of August, 1957. It should be noted, however, that whether the assumed complete reversed condition would exist awaits experimental verification. To be more specific, it means that the entrance coefficients shown in Figs. 11 and 12 in the text can be applicable to both winter heating and summer cooling conditions, but the natural draft factor and forced draft factors

† Pittsburgh Plate Glass Company.

shown in Figs. 5 and 6 may be applicable in one but not in other conditions. This was mentioned in research on infiltration in residences done at the University of Illinois†. In other words, infiltration through entrances of summer air-conditioned tall buildings may not be evaluated by the data presented in this paper.

All are aware of the fact that the leakage of the cooled air is more costly than that of the warmed air, and it is desirable that further work under summer cooling conditions be encouraged and carried out. Also, in order to better understand the characteristics of infiltration, information on pressure differences across exterior walls of buildings at various levels is greatly needed.

In regard to Mr. Parsons' discussion concerning the effectiveness of the *air curtain* in reducing infiltration, the author is inclined to think that the air curtain would be applicable only to low buildings wherein the pressure differentials existing across the entrances are small. In order to overcome big pressure differentials and high velocity of infiltration air in tall buildings, *air jet* should be provided instead of air curtain. This obviously would be objectionable to passers-by. Mr. Parsons also suggested that more work should be done on the effect of wind upon the pressure differential in isolated buildings. Observations so far have shown that wind velocity at the street level is small‡*. In one of the field tests on a building located at the bank of a river, the observed pressure differentials across the entrance at street level showed little difference with appreciable wind and with relatively no wind. Mr. Parsons also is interested in knowing about the effect of pressurizing the building in eliminating infiltration at the entrance. The effect of blower action on the pressure difference is shown in Fig. 6 of the text.

The author agrees with Professor Bahnfleth that at least 3 things could be done to control the entrance infiltration, *i.e.*,

1. sealing the entrance by using proper type of entrance doors provided it is economically and environmentally feasible;
2. reducing the pressure differential across the entrance by pressurizing the structure with ventilation air; and
3. reducing the pressure differential across the entrance by sealing or tightening other parts of the building envelope.

The author also feels that additional field tests on draft factors for newer types of structures (presumably tighter than average buildings) are very desirable and should be encouraged.

The author appreciates the remark made by Mr. Miles that tightness of the building is also an important factor affecting the entrance infiltration. It is correct that there should be no infiltration across entrances in buildings no matter how tall they are so long as their envelopes are *100% tight or sealproof*. The paper showed that a great many conventional buildings tested, which apparently are not 100% tight, have a natural draft factor of 0.7. It also is mentioned that a presumably tighter-than-average building has a lower draft factor, but data are far from enough for quantitative correlation. In acknowledging his discussion, I would like to add to the second conclusion of the paper that the most important variables determining the infiltration rate through building entrances are (a) building height (b) indoor-outdoor temperature difference (c) the quantity of air supplied and/or exhausted (d) traffic rate (e) type of building entrance and (f) tightness of building envelope. I believe that Mr. Miles would agree that until information on the criteria of tightness of building envelope and on the draft factors for buildings with various degrees of tightness is available, the last factor is only a qualitative statement, not quantitative.

---

† ASHAE RESEARCH REPORT No. 1615—Measurement of Infiltration in Two Residences, Part II—Comparison of Variables Affecting Infiltration, by D. R. Bahnfleth, T. D. Moseley and W. S. Harris (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 455).

‡ ASHAE RESEARCH REPORT No. 1615—Measurement of Infiltration in Two Residences, Part II—Comparison of Variables Affecting Infiltration, by D. R. Bahnfleth, T. D. Moseley and W. S. Harris (ASHAE TRANSACTIONS, Vol 63, 1957, p. 460, 467, 475, 476).

* Heat Requirements of Snow Melting Systems, by W. P. Chapman and Samuel Katunich (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 370, 371).

Regarding Mr. Ratai's inquiry about the flow coefficients, only those of the typical arrangements as shown in Fig. 7 of the text were calculated. Actually, with the aid of Figs. A-1, B-1, and B-2 in the Appendix, the flow coefficients of entrances with any door arrangement can be determined. However, this would not be necessary, as Figs. 11 and 12 show that the entrance coefficients for manually-operated single-bank and vestibule entrances of various arrangements do not differ appreciably for a given traffic rate, no matter how different are the traffic patterns. His other questions are concerned with the basis of the infiltration rates and construction details of the buildings. Infiltration rates shown in Fig. 13 was based on a 3- x 7-ft door area, *i.e.*, 21 sq. ft. No attempt was made to survey the construction details of the buildings in this investigation.

In closing, the author wishes to thank each of the discussers for his interest and remarks. These remarks significantly increase the value of the paper. The author is indebted to Alabama Polytechnic Institute for making this presentation possible and especially to Professor P. J. Potter for his encouragement.

1644

No. 1644

CORROSION INHIBITION ON TUBES IN LOW-PRESSURE STEEL BOILERS

By W. A. KEILBAUGH* AND F. J. POCOCK**, ALLIANCE, OHIO

THERE HAS been an increasing number of low pressure steel boilers installed in homes, apartments, hospitals, and similar buildings during recent years. Coincidental with this increase in the number of boilers installed, there has been an increase in reported tube failures attributed to waterside corrosion.

Under the direction of the Engineering Committee of the *Steel Boiler Institute*, a program is being carried out to investigate causes and prevention of corrosion in these units. It is hoped that the control methods developed by this investigation will, if practiced, lead to the practical elimination of corrosion difficulties and further strengthen the position of steel boilers in a field where they are already accepted as durable, economical heat producers.

Discussions with the Engineering Committee of the *SBI* prior to the beginning of investigation indicated that the principal cause of the corrosion difficulties was the unavoidable presence of oxygen in the feedwater and boiler water. Since deaeration of the feedwater is not practical or the addition of chemical oxygen scavengers is not generally feasible from a control standpoint in these low-pressure steel boilers, other methods of approach to the oxygen corrosion problem were considered. There appeared to be two major possible modes of attack: (1) Use of chemical inhibitors whose operating mechanism does not rely upon reaction with oxygen. It was obvious that chemical consumption by oxygen, with the consequent necessity of short interval chemical replenishment, would be an intolerable inconvenience for most boiler owners. (2) Use of corrosion resistant materials consistent with the economics of the cost of low-pressure fire tube boilers.

A major consideration in the stated approach to the problem was the requirement that if chemical inhibition were to be used, the method of inhibition should not require the services of a skilled water-treatment service engineer.

It has been apparent that corrosion testing involving the use of coupon material in laboratory bench scale tests or in autoclaves does not afford conditions compar-

---

* Head, Chemical Section, Babcock & Wilcox Co., Research Center.
** Senior Chemist, Babcock & Wilcox Co., Research Center.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

able to the actual operation of a boiler, since temperature, heat transfer and operating conditions are different.

For this reason, the Engineering Committee of *SBI* decided to study the problem of tube corrosion in laboratory operation of test boilers under conditions as close as possible to those found in actual field units.

Since *SBI* and the authors' company have a common interest in this area, the research facilities of the company were offered for the purpose of carrying out the investigation. The *SBI* accepted this offer with the stipulation that test conditions and operating procedure be subject to approval by their Engineering Committee.

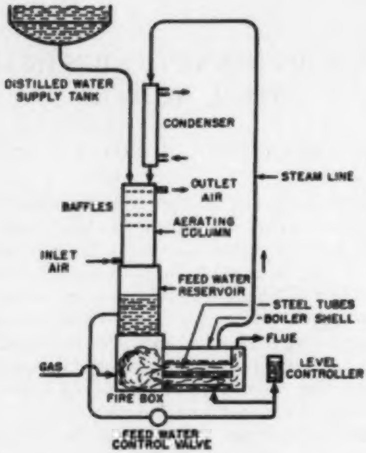


FIG. 1—SCHEMATIC DIAGRAM OF THE TEST UNIT

mittee. The program herein described was started in the latter part of 1954 and this report is a record of progress from test inception to April 1957.

BOILER DESIGN

The test equipment provided by the *SBI* for this investigation consists of 4 gas-fired steel boilers, each of which has its own feedwater and condensate system.

Each unit contains, except as noted later, four 3-in. OD by 48-in. long electric resistance welded boiler tubes made to *ASTM*-A-178 Grade A specifications which are rolled into ⅜-in. carbon steel tube sheets. Each boiler is fired with a conventional 120,000 Btu per hr gas burner.

The units operate at essentially atmospheric boiling conditions. The steam is condensed and the condensate is passed counter-currently to air flow through a baffled aerating tower. The air saturated feedwater is then fed automatically to the boiler as required by the boiler water level controls. Total steam flow for each unit is approximately 50 lb per hr. Steam losses are made up automatically

from aluminum storage tanks containing distilled water. Using distilled water as a starting point, water chemical characteristics can be varied as desired.

Automatic timing controls are provided so that the boilers are operated in pairs for 30-min firing intervals with an equivalent *off* period. This condition was

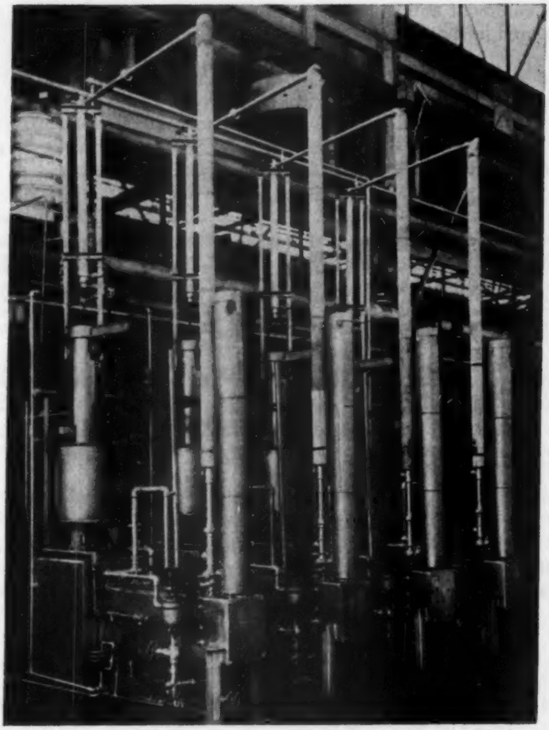


FIG. 2—GENERAL VIEW OF TEST INSTALLATION

chosen to simulate as closely as possible the periodic operation of the usual heating boiler.

The boilers are operated 24 hr a day in this manner for the duration of each test. Fig. 1 shows a schematic drawing of a typical test unit. Fig. 2 shows a general view of the test installation.

BOILER TUBES

Boiler tubes used thus far in the investigation have been electric resistance welded boiler tubes made to *ASTM*-A-178 Grade A specifications except in 1 case.

One test was operated using high strength, low alloy steel tubes. The tubes are supplied in 6-ft lengths and are cut to the 4-ft length required for the test boilers. As received, the tubes have a thin film of soluble oil as corrosion protection in transit.

The tubes are rolled into the test boilers in the *as received* condition except for light wiping with cloth and mineral spirits to remove foreign material which is picked up in shipment and some of the oil.

The tubes used in these tests were not acid cleaned (*i.e.*, pickled).

TABLE 1—TUBE METAL ANALYSES

TEST NO.	1 & 5	2	3	4	6	7, 11 & 15	8	9 & 10	12	13[b]	14 & 16
						PERCENT					
Carbon	0.17	0.12	0.12	0.10	0.11	0.17	0.07	0.08	0.12	0.08	0.15
Silicon	0.01	0.01	0.015	0.01	0.01	0.01	0.01	0.01	0.01	0.36	0.01
Manganese	0.35	0.35	0.34	0.34	0.41	0.38	0.38	0.34	0.43	0.29	0.44
Phosphorus	0.008	0.008	0.010	0.010	0.008	0.013	0.007	0.010	0.010	0.089	0.01
Sulfur	0.023	0.019	0.021	0.016	0.029	0.037	0.017	0.04	0.05	0.039	0.029
Nickel[a]	0.059	0.054	0.043	0.045	0.13	0.13	0.13	0.058	0.050	0.28	0.052
Chromium[a]	0.056	0.052	0.051	0.050	0.05	0.06	0.05	N.D.[c]	N.D.[c]	0.69	0.05
Vanadium[a]	0.005	0.005	0.005	—	—	—	—	—	—	—	—
Molybdenum[a]	0.008	0.008	0.008	0.008	0.03	0.02	0.03	0.01	0.005	0.008	0.009
Copper[a]	0.06	0.07	0.08	0.07	0.06	0.06	0.06	0.046	0.064	0.30	0.07
Aluminum[a]	0.005	0.005	0.005	0.005	—	—	—	0.001	0.002	0.005	0.004
Tin[a]	0.006	0.008	0.007	0.007	0.009	0.007	0.008	—	—	—	—

[a] Spectrographic Trace Metal Analyses.
[b] High Strength, Low Alloy Steel.
[c] N.D. = Not Detected.

Chemical Composition of the Boiler Tubes: Representative tubes from each steel heat used in this investigation were analyzed chemically. Analyses of the type A plain carbon steel tubes and the high strength, low alloy steel tubes are contained in Table 1.

BASE BOILER WATERS

Owing to the wide variations that occur in natural water supplies, it was agreed that the make-up water would be laboratory distilled water. Except for the initial test, the test solution in contact with the boiler tubes consisted of oxygen saturated distilled water containing 100 ppm chloride as sodium chloride. Sodium chloride was not added to the test solution during the first test period.

The corrosive attack suffered by the boiler tubes in this base boiler water is shown in Figs. 3 and 4.

INHIBITORS

To date, 4 inhibitors have been used in various test periods. They are as follows: (1) *A Buffered Chromate Compound.* This inhibitor is marketed under the auspices of the *SBI.* (2) *Borate-Nitrate-Nitrite Compound.* This inhibitor is a proprietary chemical compound distributed by a water treatment company for the express purpose of corrosion inhibition in diesel engine cooling systems. (3) *Sodium*

Molybdate. This material was tested on the basis of the reported work of others.† The material used was reagent grade sodium molybdate meeting *ACS* (*American Chemical Society*) specifications. (4) *Sodium Hydroxide.* Sodium hydroxide added in a concentration sufficient to produce a pH of 11.0 was used in 1 test. The sodium hydroxide used was reagent grade meeting *ACS* specifications.

TESTING TECHNIQUE

Start-up for the tests consisted of installing new tubes, as required by individual test series and as previously described, followed by a short boil out with 0.3 percent sodium hydroxide solution to remove residual oil. Following the boil out, the boilers were drained and rinsed with zeolite treated water. After the treated water rinse each boiler was rinsed with distilled water, drained, and refilled with the test boiler water. Samples of the new tubing installed in the boilers were analyzed chemically.

An operating log was maintained for routine maintenance and recording of gas pressure, cooling water pressure, boiler water level, and water analyses.

Samples of the boiler water were taken twice weekly for analysis. Usual analyses consisted of pH, chloride, conductivity and inhibitor concentrations (when used). Periodically the feedwater was analyzed for dissolved oxygen and evidences of carryover.

Each set of waterside conditions was tested for approximately 6 months. One unit was operated with uninhibited oxygenated-distilled water in an early test period to serve as a blank for comparison with the tubes from boilers to which inhibitors were added. The test conditions and test numbers are shown in Tables 2 and 3.

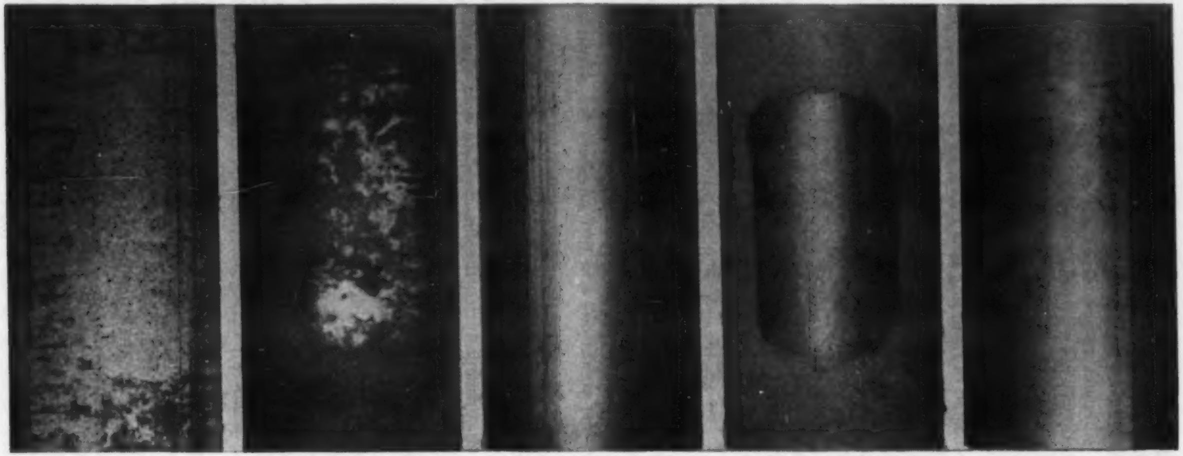
Water Samples: Boiler water samples were taken through a cooling coil from the bottom of the boiler. Feedwater samples were taken from the main feedwater line leading to the feedwater control valve. A cooling water coil was installed on the feedwater sampling line, principally for the dissolved oxygen sampling, since condensate temperatures averaged 100 F and standard dissolved oxygen analytical methods dictate a sample temperature of 65 ± 5 F.

Methods of Water Analysis: pH measurements were made with a Model H-2 pH-meter incorporating a glass-calomel electrode pair. Chloride was determined by the Mohr titration method. Special techniques for chloride analysis in waters containing the borate-nitrate-nitrite material were required. Water in this case was treated with activated charcoal to remove the organic matter, followed by pH adjustment and the usual Mohr titration for chlorides. Conductivity measurements were made with a commercial conductivity bridge using a glass *dip type* conductivity cell. Standard curves plotting chloride and chromate, chloride and borate-nitrate-nitrite, or chloride and sodium molybdate *vs* conductivity were made to maintain control concentrations of the various inhibitors. Sodium hydroxide determinations were made by *ASTM* method D-514-17 (acidimetric).

Dissolved oxygen was determined by the conventional Winkler procedure, the usual precautions being taken to avoid air contact with the sample. The dissolved oxygen concentration of the feedwater varied from 4.6 milliliters per liter to 5.0

---

† Molybdate and Tungstate as Corrosion Inhibitors and the Mechanisms of Inhibition, by W. D. Robertson (*Journal of the Electrochemical Society*, Vol. 98, No. 3, 1951).



Test No. 1-3307 hr oxygenated distilled water. Heavy generalized corrosive attack.

Test No. 2-3307 hr oxygenated distilled water. Buffered chromate added after initial corroding period. Corrosion halted.

Test No. 8-3148 hr oxygenated-chloride distilled water. Buffered chromate for duration of test. No corrosion.

Test No. 3-3307 hr oxygenated-distilled water. Borate-nitrate-nitrite concentration 7500 ppm. No corrosion.

Test Nos. 7, 11, 15-8311 hr oxygenated-chloride distilled water. Borate-nitrate-nitrite concentration reduced from 7500 ppm to 2500 ppm. No corrosion.

Note: Hours reported here do not include downtime (see Table 1)

FIG. 3—PHOTOGRAPHS OF TUBES AFTER TEST (½ DIAM)

Test No. 9-2615 hr oxygenated-chloride distilled water. Sodium molybdate added after initial corroding period. Corrosion not stopped.

Test No. 10-2675 hr oxygenated-chloride distilled water. Sodium hydroxide added after initial corroding period. Corrosion not stopped.

Test No. 14-2548 hr oxygenated-chloride distilled water. Copper tankless heating coil in contact with boiler water. Corrosion not accelerated.

Test No. 13-2548 hr oxygenated-chloride distilled water. High strength, low alloy steel tubing installed. Mild surface corrosion.

Test No. 16–2548 hr oxygenated-chloride distilled water. Physico-chemical feedwater conditioner installed. Widespread corrosion.

Note: Hours reported here do not include downtime (see Table 1)

FIG. 4—PHOTOGRAPHS OF TUBES AFTER TEST (½ DIAM)

TABLE 2—SUMMARY OF TEST CONDITIONS

Test Solution: Boiler Water No.	Test No.	Time Tubes Immersed in Boiler Water, Hours: Operating[a]	Time Tubes Immersed in Boiler Water, Hours: Down-Time	Type Tubes	Tube Condition	Inhibitors[d]
1[b]	1	3307	1656	E.R.W. *ASTM*-A-178 Grade A Boiler Tubes	NEW	NONE
	2	3307	1656	"	NEW	No inhibition initially followed by inhibitor A
	3	3307	1656	"	NEW	Inhibitor B
	4	3338	1625	"	NEW	Inhibitor A
2[c]	5	3148	788	"	Same as Test #1	Continued in test with addition inhibitor A
	6	3148	788	"	NEW	No inhibition initially followed by inhibitor B
	7	3148	788	"	NEW	Inhibitor B
	8	3148	788	"	NEW	Inhibitor A
	9	2615	865	"	NEW	Inhibitor C
	10	2615	865	"	NEW	Inhibitor D
	11	2615	865	"	Same as Test #7	Inhibitor B
	12	2615	865	"	NEW	NONE—Tubes painted and paint spattered
	13	2548	1388	E.R.W. High strength low alloy	NEW	NONE
	14	2548	1388	E.R.W. *ASTM*-A-178 Grade A	NEW	No inhibition copper tankless heating coil installed
	15	2548	1388	"	Same as Test #7	Inhibitor B
	16	2548	1388	"	NEW	No inhibition, physico-chemical water conditioner

[a] Operating time refers to actual cyclic firing of the boilers as described elsewhere. Down-time is due to inspections, routine maintenance and mechanical failures.
[b] Boiler water No. 1 is air saturated distilled water.
[c] Boiler water No. 2 is air saturated distilled water with 100 ppm chloride.
[d] Inhibitor code: (A) Buffered chromate. (B) Borate-nitrate-nitrite. (C) Sodium molybdate. (D) Sodium hydroxide.

TABLE 3—SUMMARY OF TEST RESULTS

Type Inhibitor[a]	Test No.	Inhibitor Concentration ppm (Average)	pH Average	Chloride as ppm CL (Average)	Dissolved Oxygen Conc. MLS Liter Average	Deepest Pit Measured, Inches	Number of Pits per Lineal Foot	Comments
NONE	1	..	7.6	Not Added	5.2	0.015	3300	Generalized pitting attack
A	2	2116	10.5[b]	"	5.4	0.01	840	Corrosion halted
B	3	7281	9.1	"	[c]		No Pitting	No corrosion
A	4	1860	9.8	"	5.3	0.01	25	Slight corrosion[d]
A	5	1904	10.9	99	4.9	0.016	Note (1)[e]	Corrosion halted
B	6	7183	9.5[b]	100	5.1	0.008	Note (2)[e]	Corrosion stifled
B	7	6318	9.1	99	[c]		No Pitting	No corrosion
A	8	2000	9.3	109	5.0		No Pitting	No corrosion
C	9	1978	10.5[b]	98	4.5	0.005		Some inhibition of further corrosion
D	10	..	11.0[b]	90	4.7	0.014	Note (3)[e]	Corrosion not halted
B	11	4000	9.1	92	[c]		No Pitting	No corrosion. Concentration lowered gradually
NONE (painted)	12	..	7.6	102	4.8	Etched Appearance	Not Counted	Corrosion accelerated
NONE (high str. steel)	13	..	10.0	95	5.1	0.003	615	Generalized surface brightening with slight attack
NONE (copper coil)	14	..	10.4	89	5.3	0.011	2770	Did not appear to localize or accelerate attack
B	15	2500	9.2	97	5.5		No Pitting	No corrosion. Concentration lowered gradually
NONE (Physico-chemical device)	16	..	10.1	94	4.8	0.020	2940	Did not prevent corrosion

[a] See Table 2 for inhibitor code.
[b] Average pH value after addition of the respective inhibitors.
[c] Excessive carryover experienced with Borate-nitrate-nitrite inhibitor at these concentrations prevented the chemical determination of the dissolved oxygen in the feedwater.
[d] The slight corrosion noted on these tubes is attributed to an initially low chromate concentration. Also, this boiler was used for control adjustment during shake-down period.
[e] Notes (1) No increase in incidence of pitting. (2) Compared photographically with earlier inspection. (3) Deep pits overlayed with carbuncles.

milliliters per liter for the duration of each test. This value corresponds to 90–100 percent saturation of a 100 F feedwater with air.

Corrosion Measurement: The corrosion pit depths were measured microscopically and the number of pits per unit of metal surface area were counted where possible. Microscopic examination of the tubing in place in the boilers consisted of taking a clay impression of the deepest visible pits and then measuring the height of the nodule formed on the clay block. The curved surface of the tube makes other methods extremely difficult. Data taken on the removed tubes were obtained by direct measurement of the pit depth with a microscope calibrated for this purpose. Photographic evidence was also employed for comparisons between tubes.

Discussion

Figs. 3 and 4 show graphically the effect of oxygen on boiler tubes in these tests. The tubes operated for 3307 hours (Test No. 1) in uninhibited boiler water show heavy generalized corrosion resulting from dissolved oxygen attack. Field observations have shown the 2 types of corrosion that may be experienced. One type is highly localized and intense, while the other is generalized and widespread. The second type of corrosion is less serious since the life expectancy of a boiler tube can only be gaged by the deepest pit. It was hoped that during the testing reported here, that some localized and intense pitting might occur under laboratory controlled conditions so that differentiation between the conditions causing both types of corrosion might be ascertained. Unfortunately, in the testing so far completed, no such occurrence has taken place. The corrosion experienced has been general and widespread and selective corrosion reportedly occurring in some cases as a single pit on a tube or at the tube and tube-sheet junction did not occur.

It is postulated that localized intense pits have their inception at periods of boiler storage or in areas where circulation velocities are low and localized stagnation may occur. Conditions such as these would lead to the formation of highly concentrated differential aeration corrosion cells which are well known and which have been noted many times in improperly stored industrial and power plant units.

The reference to the addition of an inhibitor after an initial corroding period refers to the fact that it was desired to obtain a proven inhibitor which could be added to existing heating boilers, many of which had already suffered some oxygen pitting. If the inhibitor was unsuccessful in halting corrosion once initiated under test conditions, then its value in existing installations would be questionable. For this reason, all the inhibitors used were subjected to this operation.

The borate-nitrate-nitrite (Tests 7, 11 and 15) inhibitor was first used at an extremely high concentration of 7500 ppm. At this concentration considerable foaming and carryover were experienced. Subsequently the inhibitor concentration was reduced in a step-wise manner over an 18-month period to test its effectiveness at lower concentrations. Below 3000 ppm, carryover was not detected in this test equipment and the tubes still showed no corrosion or incipient pitting.

Early tests with the buffered chromate inhibitor (No. 4) at concentrations of the order of 1500 ppm showed a few incipient pits. This test unit had been used to establish control settings on the test installation and the exact cause of the observed pitting was unknown. However, based on previous experience with chromate inhibitors in other applications, it was decided to use a level of 2000 ppm for further testing (No. 8). No carryover was experienced with the chromate inhibitor at this concentration level and there was no corrosion. The results of this investigation are graphically shown in Fig. 3.

Sodium molybdate (Test No. 9) was tried as an inhibitor based on bench scale testing reported in the literature. This work indicated successful inhibition with the following chemicals: sodium molybdate, sodium tungstate, sodium chromate, and sodium nitrite. Coupon type bench tests in the laboratory screening work with sodium molybdate at a temperature of 200 F in air saturated solution showed good results. It was agreed that this test should be carried out in the model boilers. Results of one test were not satisfactory (Fig. 4) and the application of sodium molybdate for the purpose of inhibition under these test conditions was discontinued.

A test involving the use of sodium hydroxide (No. 10) to produce a boiler water pH of 11.0 was carried out. It was felt by some that the use of sodium hydroxide at a pH of 11.0 would be sufficient to deter further heavy corrosion. The results are shown in Fig. 4. It is evident that sodium hydroxide alone is not sufficient. While the severity of the oxygen attack may be diminished through increased pH, this type of approach is not an effective means of protection.

In order to illustrate that the inadvertent contamination of boiler tube surfaces by paint during boiler manufacture has a deleterious effect on operational tube life, one test (No. 12) involving this consideration was carried out.

Two of the tubes in this test had their entire waterside surface covered with a typical paint used for external boiler protection. Two other tubes were simply spattered with aluminum paint. These tubes were operated in uninhibited boiler water. The results of this test showed that corrosion was accelerated by the paint contamination.

Low alloy steels have been considered for use in low pressure steel boilers. It was stipulated however, that the material used be consistent with the economics of the cost of low pressure steel tube boilers. One test (No. 13) involving the use of high strength, low alloy tubing in chloride oxygenated distilled water without inhibition has been carried out thus far. Fig. 4 shows the condition of this tubing after test. Pits were few in number and shallow; the type of attack was general in nature and not so severe as has been observed in other cases with plain carbon steel tubes. It should be pointed out that the analysis of this steel varies from heat to heat, probably more so than plain carbon steel. However, the work of others‡ indicated that the copper content of the steel is related to corrosion resistance in sea waters. These investigators also illustrated that this same condition does not appear to hold true at higher alkalinities. Also, apparently above about 0.2 percent copper, no additional advantage is gained.

With the thought that copper, a dissimilar metal, might adversely affect tube corrosion resistance, an investigation was made of the influence of a copper tankless heating coil installed in contact with the boiler water (Test No. 14). The total surface area of the copper tankless heating coil immersed in the boiler water was approximately 3 percent of the ferrous boiler surfaces in contact with the boiler water. This test was carried out with plain carbon steel tubing in uninhibited base boiler water. The results shown in Fig. 4 reflect the examination of these tubes which showed no localization or acceleration of corrosion during the test period.

In view of widespread attempts to sell physico-chemical water conditioning devices to the industry, it was decided to test one of these units (Test No. 16).

---

‡ Model Boiler Tests on the Influence of the Copper Content of the Steel on the Corrosion of Tubes in Artificial Sea Water, by G. M. A. Butler and H. C. K. Ison (*The Institute of Marine Engineers*, February 12, 1957).

The water conditioning device used was of a type that is installed on the boiler feedline with normal piping connections. No power source or other connections are used. This unit was installed on the boiler feedline under conditions as directed by the manufacturer and the boiler was operated with uninhibited test boiler water for one test period. For this single test the device was unsuccessful. (See Fig. 4).

It should be pointed out here that the tests so far performed neglect the effect of hardness in the water. It is felt that the presence of calcium or magnesium ions may have an effect on the required inhibitor concentrations and for this reason, tests are now in operation incorporating the addition of hardness to the boiler water.

Field experiences in the boiler industry have indicated that boiler storage conditions are closely related to extensive corrosive damage involving boiler tubes. It has been found that some localized corrosion has its inception during periods of lay-up and extensive downtime when the boiler is wet or contains a moist atmosphere. Accordingly, tests are being carried out under static conditions to investigate corrosion during storage and the efficiency of inhibitors in preventing attack during these periods. In these tests, small carbon steel boxes have been constructed with simulated tube sheets forming two of the sides. Plain carbon steel tubes were rolled into the tube sheets and these boxes have been laid up at atmospheric conditions for 3-month intervals. These tests are incomplete at this time.

CONCLUSIONS

It is readily apparent that dissolved oxygen will cause immediate and widespread corrosive attack under these test conditions. This attack is best explained by the electrochemical theory of corrosion and is well documented.

Testing apparatus and technique have been developed which more nearly duplicate operating conditions for low pressure heating boilers than do bench type coupon or autoclave types of corrosion and inhibitor investigations.

Test operation to date has produced the following conclusive results for the previously described test conditions.

1. The buffered chromate type inhibitor is effective in halting and preventing corrosion caused by dissolved oxygen under operating test conditions. The minimum effective inhibitor concentration appears to be 2000 ppm.
2. The borate-nitrate-nitrite inhibitor is effective in preventing corrosion caused by dissolved oxygen under these test conditions. The minimum concentration appears to be 3000 ppm. At this concentration, carryover is not observed in the test equipment.
3. Sodium molybdate and sodium hydroxide are not effective in halting dissolved oxygen attack in the test boilers.
4. The immersion of a copper tankless heating coil in the boiler water, out of contact with the tubes, does not accelerate or localize corrosive attack under these test conditions.
5. The physico-chemical feedwater conditioner used in this test program was ineffective in preventing or reducing corrosive attack by dissolved oxygen.
6. One test with high strength, low alloy steel tubing indicated a less severe attack than was observed on carbon steel tubing under the same test conditions. Some attack was noted but the pits were shallower and the incidence of pitting was lower.

PRESENT AND FUTURE ACTIVITY

At present, the buffered chromate and the borate-nitrate-nitrite inhibitors are being investigated in the presence of calcium hardness in the water. It is prob-

able that magnesium hardness will also be included at a later date. One other high strength low alloy steel is under test and investigation of influence of boiler storage conditions is continuing.

It is believed that it should be possible in the not too distant future for the *SBI* to originate a revised set of instructions for the care of the water side of the low pressure units.

ACKNOWLEDGMENTS

This research is being carried out by the Research and Development Center of the Babcock & Wilcox Co., Tubular Products Division, for the *Steel Boiler Institute.*

The guidance of the Engineering Committee of *SBI* continues to be of utmost importance in furthering the objectives of the test program.

The consultation and help given to the writers by H. F. Hinst, chief metallurgist, Keystone Plant of The Tubular Products Division, Babcock & Wilcox Co., is also gratefully acknowledged.

BIBLIOGRAPHY

Corrosion in Scotch Marine Boilers, Part I, by F. D. Wormell, G. M. A. Butler, and J. G. Beynon (*The Institute of Marine Engineers Transactions*, Vol. 69, April 1957, p. 109).

DISCUSSION

F. N. SPELLER, Pittsburgh, Pa. (WRITTEN): The tests reported on inhibition of corrosion in low pressure boilers point to 2 anodic inhibitors that are effective under the test conditions with basic distilled water.

Chromate treatment is the most powerful, but is subject to reduction by organic matter and other soluble matter, so that a simple test for chromate concentration that should be applied regularly is necessary. A quick color test can be used for this purpose.

The *nitrite* inhibitor may prove more generally reliable but it would be well to await results of further research before such inhibitors are generally used, as all anodic inhibitors are apt to be dangerous and accelerate pitting when used below a certain concentration.

L. N. HUNTER, Johnstown, Penna. (WRITTEN): I want to express my appreciation to the authors of this paper and the company by whom they are employed for the tremendous amount of work and expense that has been involved in developing the information for this paper. It represents a real contribution to the heating industry.

It has been my privilege to be associated with this program as Chairman of the Engineering Committee of the *Steel Boiler Institute* since the inception of the program a number of years ago. In addition to the technical information given in the paper, the background that preceded the start of this research program is interesting.

Why was the program started?

All fuel burning equipment is susceptible to corrosion of one form or another and the nature of the corrosion depends, to a certain extent, on the nature of the equipment. Steel boilers have been no exception. Even though corrosion problems have not been limited to steel boilers, the steel boiler industry for many years was not frank in admitting it existed, probably because of fear that it would react unfavorably on the usage of steel boilers.

A few years ago this thinking changed and members of the *SBI* felt that it should be admitted that steel boilers that do not receive proper attention are susceptible to cor-

rosion and furthermore that we should develop the necessary information to prove that this corrosion can be avoided.

The corrosion problems in low pressure heating boilers, which are those usually used for heating service, can be more serious than those in large power boilers for the simple reason that although the service is likely to be more severe in the case of power boilers they are also much more likely to receive individual and continuous attention. They are cleaned more frequently and water treatment is frequently provided.

There are several reasons why boilers can corrode, but the reason discussed in the paper, namely oxygen corrosion, is probably responsible for well over 95 percent of failures due to corrosion today.

Most of the boilers used today are automatically fired and the fuels are relatively clean. It is very seldom, therefore, that one finds a boiler failing due to corrosion from the fire side. I am not inferring that it does not exist, but its occurrence is infrequent. Certainly if the heating surfaces are kept reasonably clean, the danger of fireside corrosion is practically eliminated. The boiler must be designed so the heating surfaces can be readily cleaned and there should not be an opportunity for deposits of soot to collect in inaccessible places.

The majority of failures that occur today are due to oxygen corrosion attacking the tubes of the boilers and I believe the authors have demonstrated that if the oxygen is kept to a relatively low level there should be no difficulty with corrosion. In addition to the use of the proper kind of compound in the boiler, there are a number of other steps that can be taken to discourage corrosion.

Many boilers are equipped with domestic water heaters which require that the boiler water be kept hot winter and summer to supply the domestic hot water and keeping the boiler hot reduces corrosion on both the water and fire sides of the boiler.

Boilers should not be drained and refilled if it can be avoided and if the boiler is refilled with fresh water it should be heated immediately to 180 F or more for a period of time to drive off the dissolved oxygen. Filling a boiler with fresh water and allowing to stand for several weeks or even several days can start corrosion, depending upon the nature of the water.

We sometimes hear of identical boilers installed side-by-side where one will fail after a period of time due to corrosion and the other one will show no evidence whatsoever of corrosion. It is quite possible that a simple matter like filling one boiler with water and allowing it to stand could be responsible for this kind of condition occurring.

When corrosion does occur, the owner of the boiler or the operator very often jumps to the conclusion that the material of construction must be defective. If the boilers are constructed in accordance with the *ASME* Boiler Construction Code, one can be pretty sure that the materials of construction or the workmanship are not to blame.

E. M. MITTENDORFF, Charlottesville Va.: I would also like to commend the authors on the work that they have done in the study of corrosion in low pressure steel boilers. Unfortunately I was not able to read this paper prior to this session and any comments that I have might be answered in the paper. However, I notice that Mr. Keilbaugh expressed the fact that the boilers were not acid cleaned before the tests were run, and wonder whether he means to indicate that, if the boilers had had an acid cleaning, that a different result might have been obtained.

A second question is whether this chromate and other inhibitors were put into the boiler in a batch or by continuous feeding.

And lastly, if any attempt was made to determine the effect of the inhibitors on such items or equipment as boiler feed control valves or boiler pop valves, or on control valves in the system.

JOHN W. JAMES, Chicago, Ill.: The authors refer to a test using a physico-chemical water-conditioning device. Over a period of about a year we have been observing the operation of one of these—I am not sure it is the same device Mr. Keilbaugh used—in a hospital in the Chicago area, where a considerable amount of replacement water is

being used from the Lake Michigan source. It appears as we observe the operation of this boiler and watch its blowdown characteristics, that this physico-chemical device tends to deposit an additional and unusual amount of particles out of the water. As a result more than the usual amount of blowdown is required. I wonder whether in the authors' tests, conducted under rather critical conditions, they noticed in the use of this device any unusual amount of deposition of solids which would lead to the conclusion that maybe if such a device is used, discounting the questionability of its corrosion advantage of control, there might be a suggestion that one ought to blow the boiler down more frequently.

The second thing I have in mind is that since there are proponents of a control arrangement which induces an electrical current to control corrosion, would Mr. Keilbaugh care to comment on such devices in the same light that he mentioned these physico-chemical arrangements?

K. T. DAVIS, Syracuse, N. Y.: I have two questions to put to the author. First, in the case of the copper coil immersed in the boiler water, was the coil electrically bonded to the shell of the boiler? And second, the paper reports only on fire tubes—none of which pass through the water line. I wonder if any observations were made of the shell of the boiler to determine the effectiveness of the inhibitor at the water line level; also, if the authors would comment on protection afforded the boiler shell and fire tubes which might pass through the water line. The paper is very interesting and well worthwhile; but I'm particularly pleased to hear that work is to go further toward determining what corrosion control can be expected when typical impurity concentrations exist in the boiler water.

R. H. SAVAGE‡, Los Angeles, Calif.: I believe that Shepard Powell, in one of his books, recites the cases of steam boilers being used in homes and buildings, without corrosion, for as long as 70 years. The reason for no corrosion was that the systems were tight and no oxygen entered the boilers after they were originally filled with water. Isn't the approach to this problem of corrosion in small systems, particularly homes, one of preventing the entrance of the oxygen? Once the boiler is filled, the oxygen that is in that water—once it reacts with the metal—is no longer available for further corrosion. We find no corrosion in hot water heating systems in Los Angeles unless there are water losses and new oxygen is allowed to enter the system with the replacement water. Now the same should be the case with steam boilers.

L. E. SEELEY, Westfield, Mass.: I asked Mr. Hunter a question privately and he suggested I ask the author. The question is: What inspection, if any, was made on the shell of the boiler during these tests?

AUTHORS' CLOSURE (Mr. Keilbaugh): The authors wish to express their thanks to Mr. Hunter for his well prepared and informative discussion. His experience and that of the *Steel Boiler Institute* Engineering Committee have provided excellent guidance for the entire test program.

Regarding Mr. Speller's remarks, it has been our thought that the amount of organic material that is found in the average water used to fill these units when they start up is not very great, particularly in relation to the rather overwhelming concentration of inhibitor that is present. We would not look for too much reduction of the chromate type inhibitor. It is believed that everything that can be done to eliminate the necessity for testing should be done, because long experience has shown that testing of boiler water is an unpleasant little task for a lot of people to do on a regular basis. It is possible that final recommendations might include draining the boiler and adding a fresh charge of inhibitor once a year or once every 2 or 3 years. However, the authors are not prepared to give exact recommendations at this time. It is felt that the nitrate-

‡ Water Chemists, Inc.

nitrite type of inhibitor offers some promise. Further testing is in progress and it is expected that the answer to Mr. Speller's precautionary note will be forthcoming.

Concerning the subject of acid cleaning as mentioned by Mr. Mittendorf, we would not expect different results if the test boilers had been acid cleaned. The chief reason for not acid cleaning was simply to approximate as closely as possible the type of treatment these units would receive when they are installed for regular field service.

The inhibitors were put into the boilers by batch feeding as required for the maintenance of the desired boiler water inhibitor concentration.

The purpose of the test program has been to produce a chemical inhibitor for protection of water-side boiler surfaces and no consideration has been given to feed valves, pressure relief valves, or other appurtenances. It would be expected that the only adverse effect on pressure relief valves would result from boiler water foaming conditions, since these are steam units in which no water should be present in the steam space where these valves are normally installed. No foaming has been apparent with the inhibitors used in those concentrations judged adequate for corrosion protection.

Under the projected plan for using the inhibitors, it is believed that there should be little or no opportunity for contact between the boiler water chemicals and feed or control valves in the system. If contact were unavoidable, no corrosion of normally used structural materials would be expected. In the event of water leakage around valve stems and similar places there would be an accumulation of inhibitor chemical salts at the leakage point where water evaporation occurred. This accumulation of chemical salts could be expected to produce *stickiness* in the movement of mechanical parts.

In reply to Mr. James question, no deposition was noticed in the use of the physico-chemical water treating device. None really should have been expected because there were no scale forming materials present in the water. The purpose in this test was to investigate one of the many claims made for the equipment, *i.e.*, it will prevent corrosion. As shown in the test, the device did not produce the results claimed by its manufacturer or its supplier. We would rather not comment on the supposed principle of operation of this treatment equipment. As far as is now known, the *SBI* has no intention of recommending further testing of any other similar devices.

Mr. Davis asked two interesting questions. In reply to the one, the copper coil immersed in the boiler water was bonded to the shell. Concerning the other, observations have been made of the effectiveness of the inhibitors in the prevention of corrosion of the test boiler shells. These observations have shown that when the inhibitors are used, there is no water line corrosion and that when the inhibitors are not used, a pitting type attack occurs just below the water line.

It would be expected that fire tubes of these low pressure units which might pass through the water line would be satisfactorily protected.

Answering the question of Mr. Savage, if one can be completely certain that oxygen will always be absent, corrosion difficulties should be nil in the average situation. However, as was mentioned in this paper, it has been judged that the long term absence of oxygen cannot be guaranteed in most installations.

1645

No. 1645

HEAT GAIN THROUGH WINDOWS SHADED BY CANVAS AWNINGS

By NECATI OZISIK* AND L. F. SCHUTRUM**, CLEVELAND, OHIO

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Avenue, Cleveland 3, Ohio.

RESEARCH on the performance of various kinds of awnings in reducing heat gain through windows has been carried out at the ASHAE Research Laboratory under the guidance of the TAC on Heat Transfer through Fenestration† as a part of a continuing program of research to develop design data for heat gain through windows. The awnings to be tested were selected at a meeting attended by representatives of the awning industry, members of the TAC, and members of the ASHAE Research Laboratory staff, and were judged to be typical for the majority of applications. The shape, color, venting characteristics, and specific material of the awning were recognized as the most important factors, and the investigation was directed to determine the relative importance of each of these variables. The percent of window coverage was kept constant in all tests.

TEST APPARATUS

The Calorimeter: Heat gain measurements were made by means of the solar calorimeter essentially as described in an earlier published paper.[1] A general view of the calorimeter with the canvas awning in position is shown in Fig. 1. The apparatus could be rotated to any desired orientation and could be tilted through 90 deg about a horizontal axis. The test window had no setback, and the surface area was 44¼ x 44¼ in. in size. A water and ethylene glycol mixture, the inlet

---

* Research Engineer, ASHAE Research Laboratory.
** Research Supervisor, ASHAE Research Laboratory. Member of ASHAE.
† Personnel: R. W. McKinley, Chairman; R. C. Jordan, Vice Chairman; W. J. Arner, T. C. Carson, E. W. Conover, R. B. Crepps, W. B. Ewing, J. E. Frazier, H. F. Kingsbury, J. B. Leavy, Manny Levine, E. C. Miles, Bruno Morabito, F. W. Mowrey, D. J. Vild, H. B. Vincent, O. F. Wenzler, C. J. Youngblood, Jr.
[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

temperature of which could be controlled as desired, was circulated through the tubes inside the calorimeter to absorb the heat gain.

Instrumentation: All temperature measurements were made by means of copper-constantan thermocouples, which permitted taking individual as well as parallel readings to obtain average temperatures.

Measurement of solar radiation was made with 2 Eppley thermo-electric type pyrheliometers, one of which was mounted on the upper face of the panel just

FIG. 1—VIEW OF CALORIMETER WITH AWNING IN PLACE

above the awning, and the other on the lower face at about 1 ft below the lower edge of the test window. A third pyrheliometer was mounted on the same surface of the panel at the same level as the lower edge of the awning, in order to take measurements of the ground reflected component of solar radiation.

Low temperature radiation received from the sky and the surroundings was measured by a radiometer described in Reference 2.

Wind velocity was measured by a calibrated cup-type anemometer. Two 16-point electronic recorders provided continuous readings throughout the tests.

Awnings and Glasses: The four different awnings selected for testing included:

1. Conventional type, canvas, outside dark green, underside grey-green.
2. Conventional type, canvas, outside white, underside grey.
3. Venetian type, canvas, outside dark green, underside grey-green.

4. Conventional type plastic fabric (woven from polyvinylidene chloride monofiliments) outside and inside dark green.

The shape and size of these awnings are given in Fig. 2. The effect of venting the conventional type of canvas awnings was investigated by using standard head-rod clamps and extended head-rod clamps as the positioning devices. The former clamp permitted almost no venting at the top, whereas the latter allowed a ¾-in. opening at the top to vent the hot air trapped under the awning.

The canvas material was opaque to solar radiation, whereas the loosely woven plastic material permitted a fraction of the solar radiation to pass through it. The

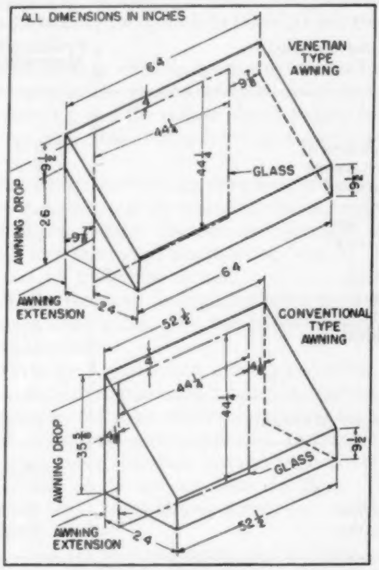


FIG. 2—SHAPE AND SIZE OF AWNING TESTED

quantity of solar radiation passing through the plastic material and falling upon the glass surface was largely a function of the angle of the solar beam with the awning surface. Variation in the transmittance of the plastic material with the incidence angle of an incandescent light beam was determined by pyrheliometer measurements. The transmittance, reflectance, and absorptance of the awning materials obtained from spectral tests and from experiments made at the Laboratory using a pyrheliometer, are shown in Table 1.

The glasses used in combination with the awnings were ¼-in. thick regular plate glass and ¼-in. thick heat absorbing glass.

Test Procedure: Tests were made with the calorimeter in a vertical position for fixed orientation or following the sun. During each of the tests of 20 min duration,

data recorded included the altitude of the sun, wall solar azimuth, wind velocity and direction, and condition of the sky in addition to the measurements of temperature and solar intensity. The ground-reflected component of the solar radiation was recorded by the third pyrheliometer which was covered on top in order to shade the pyrheliometer from the solar radiation coming from the sky. The total heat gain of the calorimeter was obtained by computation from the quantity of circulating liquid and its temperature rise, and was corrected for the heat ex-

TABLE 1—SOLAR REFLECTANCE AND TRANSMITTANCE OF AWNING MATERIALS

AWNING MATERIAL	NORMAL REFLECTANCE	
	PYRHELIOMETER[a] MEASUREMENTS	SPECTRAL[b] MEASUREMENTS 0.3—2.1 MICRON
CANVAS		
Dark green		
Outer surface (dark green)	0.21	0.22
Inner surface (grey green)	0.32	...
Silver blue (outer surface)	0.27	0.25
Green (outer surface)	...	0.36
Dusty rose (outer surface)	...	0.37
Silver rose (outer surface)	0.41	0.40
White		
Outer surface (white)	...	0.91[c]
Inner surface (grey)	0.53	...
PLASTIC		
Dark green (both sides)	0.27	...
Blue (both sides)	0.32	...
Red (both sides)	0.34	...

AWNING MATERIAL	TRANSMITTANCE
CANVAS	Zero
PLASTIC	
Direct solar radiation	0.25 normal
Diffuse solar radiation	0.15

[a] Reflectances compared with white sample in sunlight.
[b] Spectral values from Electrical Testing Laboratories, Inc. (0.45–1.25 micron) extended to (0.3 to 2.1 micron).
[c] For design data 0.70 was used to allow for weathering.

changes at the back and side surfaces of the apparatus. The total heat gain thus obtained was the sum of the solar energy transmitted through the glass and the convection-radiation gain from the glass. The transmitted solar energy was computed by subtracting from the total heat gain, the convection-radiation gain, which was obtained from calibration curves based on the temperature of the glass and the temperature of the heat absorbing surfaces of the calorimeter.

ANALYSIS OF PROBLEM

Heat Transfer through Windows: A window receives direct solar radiation from the sun, diffuse solar radiation from the sky, reflected solar radiation from the

surroundings, and low-temperature radiation both from the sky and the surroundings. If there is no shade of any kind on the window to prevent the sun's direct beam falling upon the glass, a large fraction of direct, diffuse, and reflected solar radiation passes directly through the glass into the room, a smaller fraction is reflected back into the atmosphere, and the remainder is absorbed by the glass. The fraction of solar radiation which passes directly through the glass is hereinafter referred to as the transmitted heat gain. The solar energy absorbed by the glass causes an increase in glass temperature until an equilibrium is reached between the rate of heat absorption by the glass and the rate of heat dissipation from the glass by convection and radiation, both into the room and to the outdoors, with the heat storage in the glass remaining constant at equilibrium. The heat dissipation from the warm glass into the room is hereinafter referred to as the convection-radiation heat gain.

In the presence of an awning which is opaque to solar radiation, if no direct beam falls upon the glass surface, the ground reflected solar radiation entering through the openings of the awning and the diffuse sky radiation from that portion of the sky visible by the glass under the awning form the major portion of the transmitted heat gain.

If the awning has a high surface temperature, it tends to increase the glass temperature both by radiation and by warming the air under the awning. Therefore, in the presence of the awning, the glass temperature, which controls the convection-radiation gain into the room, is a function both of the awning temperature and the temperature of air under the awning.

The total heat gain into the room through a window is the sum of the transmitted and convected-radiated heat gains. In the following sections, these two components are treated separately.

Transmitted Solar Energy: Consider an awning assembly, in which the awning shades all the glass surface so that no direct beam strikes the glass. Assuming the awning material is opaque, the solar radiation received by the glass is due to the ground-reflected and sky-diffuse solar radiation entering through the openings of the awning and falling upon the glass surface, both directly and after being reflected from the underside of the awning onto the glass. Therefore, for a given awning, the transmitted heat gain is largely a function of the ground reflected component of solar radiation.

A relation between the transmitted heat gain and the ground reflected component of solar radiation was obtained by plotting the transmittance* for awning-glass combinations against (I_{GV}/I_{dV}), where I_{GV} and I_{dV} are the intensity of the ground reflected and the total diffuse solar radiation falling upon a vertical wall having the same orientation as the window. Figs. 3 and 4 show this relation for the conventional and venetian type of canvas awnings respectively in combination with regular plate glass. The solid lines in the figure represent the calculated values, the treatment of which is given in Appendix A. The experimental values fall a little above the calculated line. This is attributed to the fluttering of the side and front flaps of the awning, which exposed a larger opening for the solar radiation falling upon the glass surface.

The canvas material was opaque to solar radiation, whereas, loosely woven plastic material permitted some solar radiation to pass through it. In this case, since the transmittance for awning and glass combinations are dependent upon both (I_{GV}/I_{dV})

---

* Solar energy transmitted per unit area of the glass surface expressed as fraction of the intensity of total diffuse solar radiation falling on a vertical wall having the same orientation as the window.

and the amount of solar radiation passing through the plastic material, the experimental results could not be expressed in a plot similar to those given in Figs. 3 and 4. However, the additional amount of solar radiation passing through the plastic material and falling upon the glass surface could be computed from the test data.

The foreground immediately in front of the calorimeter was a dark colored platform surrounded by a grass lawn. The effect of variation in foreground on the heat transmitted through the window was investigated by placing a 4 x 8 ft diffusely reflecting white surface in front of the calorimeter. When the sun was not shining on the white surface, its presence made little difference. However, when the direct rays of the sun fell on the area, the transmitted energy was 2 or 3 times as great with the white surface as with the normal foreground.

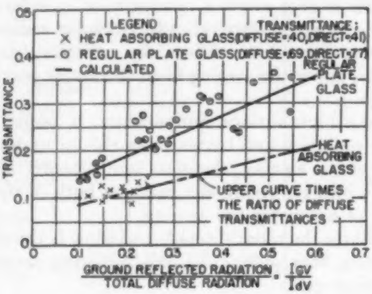


Fig. 3—Transmittance of Conventional Type Awning in Combination with Regular Plate and Heat Absorbing Glass

In order to present design data on the transmitted component of the total heat gain, the Heating Ventilating Air Conditioning Guide values for the solar radiation intensities for a clear atmosphere at 40 deg North Latitude on August 1 were taken as the basis of design-data calculations. Utilizing the relation given in Figs. 3 and 4 (See Appendix B), transmitted heat gains were calculated for conventional and venetian type canvas awnings and conventional type plastic awnings in combination with regular plate glass. These data are tabulated in Table 2A for East, South, and West orientations and for various hours of the day during which the awning prevents the direct sun from falling upon the glass. It is, however, to be noted, that transmitted heat gains given in this table are for a rather dark foreground. For a light foreground, the transmitted heat gains for canvas awnings will be about twice those given in Table 2A. For the conventional type plastic awning, an amount equal to the transmitted heat gain for the conventional type canvas awning with a dark foreground should be added.

Convection and Radiation Heat Gain: By writing the basic heat balance equations for the glass and awning, as shown in Appendix C, the convection-radiation heat gain from the glass into the room was related to the outside air temperature and the amount of heat absorbed by the awning and glass. The convection-radiation

heat gain obtained from this relation for a 75 F indoor temperature was plotted in Fig. 5 against a term which is the sum of the outdoor air temperature, one third of the solar heat absorbed per unit area of the glass, and a fraction (determined experimentally) of the solar heat absorbed per unit area of the awning. For wind velocities of 2½ to 5 mph, this fraction was approximately 0.10 for conventional type canvas awning with standard-head-rod clamp, 0.07 for the conventional type canvas awning with extended-head-rod clamp, and 0.05 for the venetian type of awning. For wind velocities above 5 mph, the value was 0.05 for all awnings.

The experimental data for the convection-radiation heat gains, after being adjusted to a 75 F indoor temperature, are in satisfactory agreement with the calculated curve as shown in Fig. 5. Values from this curve were used for calculating

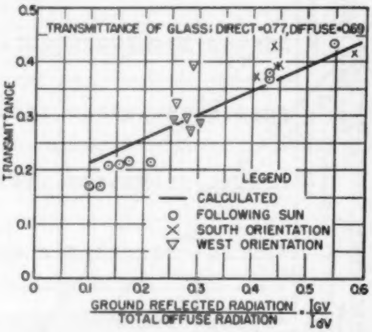


FIG. 4—TRANSMITTANCE OF VENETIAN TYPE AWNING WITH REGULAR PLATE GLASS

the design data for convection-radiation heat gains. For these design-data calculations, the outdoor air temperatures and the solar radiation intensities were taken the same as those given in THE GUIDE for a clear atmosphere at 40 deg north latitude on August 1. The convection-radiation design data thus calculated for the awnings in combination with regular plate glass are given in Table 2B, for East, South, and West orientations and for various hours of the day.

Application of Data: The total heat gain through a window shaded by an awning is the sum of the transmitted and the convected-radiated heat gains. The total heat gains thus obtained, for the window and awning combinations tested, are given in Table 2C. These values are considered to be correct to within ±2 Btu per (hr) (sq ft).

It should be noted that the heat gain values in Table 2C are strictly applicable only to the glass-awning combinations tested. The test window was 44¼ in. square, and all of the test awnings had drops* of approximately 70 percent. How-

* Fraction of the height of the window covered by the awning (see Fig. 2).

TABLE 2—DESIGN DATA ON HEAT GAIN[a] THROUGH AWNING-GLASS COMBINATIONS FOR 75F INDOOR TEMP.

Orientation	Sun Time	Outdoor Temp., F	Table—2A: Heat Gains With Awnings, Transmitted, Conventional Canvas	Table—2A: Conventional Plastic	Table—2A: Venetian Canvas	Table—2B: Heat Gains With Awnings, Convection—Radiation, Conventional Canvas Green, Standard	Table—2B: Conventional Canvas Green, Extended	Table—2B: Conventional Canvas White, Extended	Table—2B: Conventional Plastic Green, Extended	Table—2B: Venetian Canvas Green, Extended	Table—2C: Heat Gains With Awnings, Total Gain, Conventional Canvas Green, Standard	Table—2C: Conventional Canvas Green, Extended	Table—2C: Conventional Canvas White, Extended	Table—2C: Conventional Plastic Green, Extended	Table—2C: Venetian Canvas Green, Extended	Table—2D: Total Heat Gain Glass Only, Reg. Plate (b)	Table—2D: Ord. Window (c)	Table—2D: Heat Absorb. (d)
EAST	7AM	75	6+9[e]	31+9[e]	9+8[e]	13	9	4	8	8	28	24	19	48	25	183	199	135
	8	77	8	38	11	17	13	6	12	12	25	21	14	50	23	192	211	142
	9	80	9	37	11	21	17	9	16	15	30	26	18	53	26	172	187	129
	10	83	9	30	12	24	19	12	18	17	33	28	21	48	29	124	137	94
	11	87	10	22	12	26	22	16	20	20	36	32	26	42	32	67	71	54
	12	90	9	13	10	26	22	18	21	21	35	31	27	34	31	32	33	27
SOUTH	8AM	77	6	10	7+4[e]	10	10	4	7	7	16	16	10	17	18	19	20	14
	9	80	7	16	9+6[e]	16	13	8	10	11	23	20	15	26	26	44	48	34
	10	83	9	21	11+2[e]	21	18	12	16	16	30	27	21	37	29	73	78	57
	11	87	9	24	11	27	23	16	21	21	36	32	25	45	32	96	104	75
	12	90	10	29	12	30	26	19	25	25	40	36	29	54	37	107	115	84
	1PM	93	9	24	11	34	30	23	28	28	43	39	32	52	39	103	110	82
	2	94	9	21	11+2[e]	34	30	23	28	28	43	39	32	49	41	85	90	69
	3	95	7	16	9+6[e]	33	29	24	26	27	40	36	31	42	42	59	63	50
	4	94	6	10	7+4[e]	28	26	21	24	24	34	32	27	34	35	36	37	32
WEST	12	90	9	13	10	26	23	18	21	21	35	32	27	34	31	32	33	27
	1PM	93	10	22	12	33	29	22	27	27	43	39	32	49	39	73	77	61
	2	94	9	30	12	37	32	24	30	29	46	41	33	60	41	136	147	107
	3	95	9	37	11	39	34	25	32	31	48	43	34	69	42	189	204	148
	4	94	8	38	11	36	32	24	31	30	44	40	32	69	41	210	228	164
	5	93	6+9[e]	31+9[e]	9+8[e]	33	29	22	28	27	48	44	37	68	44	202	217	157

NOTE: Data obtained from a 44 in. square regular plate glass window facing a dark foreground and shaded by an awning having a drop of 70 percent. Values calculated for a combined outside conductance of 3 Btu per (hr) (sq ft) (F deg).

[a] All heat gains are in Btu per (hr) (sq ft).

[b] Regular plate glass, with transmittance for direct-solar radiation (normal) 0.77.

[c] Ordinary window glass, with transmittance for direct-solar radiation (normal) 0.87.

[d] Heat absorbing glass, with transmittance for direct-solar radiation (normal) 0.41.

[e] Addition due to direct sun on the glass.

ever, by making the corrections described later, the values in Table 2 may be applied with reasonable accuracy to other awning-shaded windows.

Within reasonable limits, the width of a window does not appreciably alter the total heat gain values given in Table 2C.

As just indicated, Table 2 has been prepared for a window shaded by an awning having a 70 percent drop. If the awning drop is less than 70 percent, sunlit glass

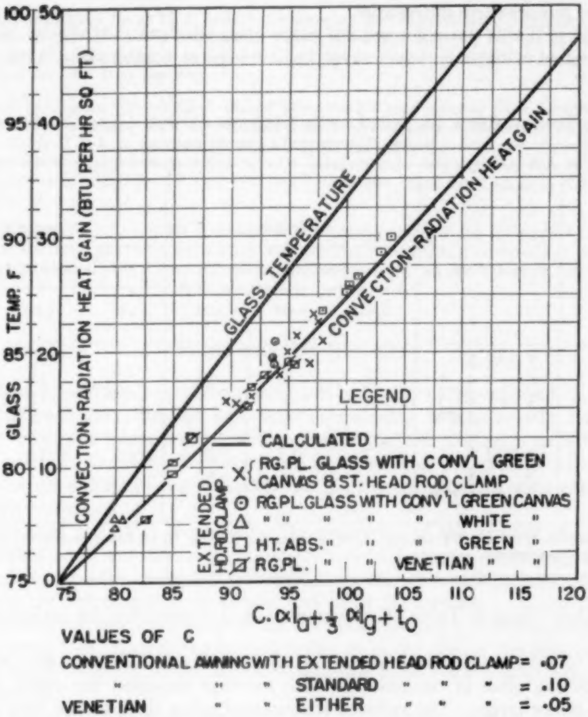


FIG. 5—CONVECTION-RADIATION HEAT GAIN AND GLASS TEMPERATURE (FOR 75 F INDOOR TEMPERATURE)

areas may exist in the lower section of the window, which have not been accounted for in Table 2. The height of this *lower section* is equal to the total glass height -1.4 x awning drop in feet. The sunlit area, if any, in the *lower section* can be estimated from the data in Table 3. This table gives the sunlit area in the lower section of a 3 x 5 ft window for 3 orientations, 3 different awning drops, and for various times of the day. The data may be applied to windows of other dimensions without serious error.

If the foreground is of light color, the total heat gain in Table 2C should be increased by an amount approximately equal to the transmitted component of heat gain given in Table 2A for the type of canvas awning under consideration. The additional gain for a plastic awning is the same as for a similar canvas awning.

The data in Tables 2B and 2C are for an inside temperature of 75 F, and for outside temperatures as indicated in the table for different hours of the day. For temperature differentials other than those used in the table, the data may be corrected by adding or subtracting 1 Btu per (hr) (sq ft) for each degree of increase or decrease in temperature differential.

The data in Table 2 are for regular plate glass windows. However, the values can be applied with only small error to awnings in combination with ordinary

TABLE 3—SUNLIT AREA ON THE LOWER-SECTION OF THE GLASS AS PERCENTAGE OF THE ENTIRE GLASS SURFACE

Orientation	Sun Time	Conventional Awning Drop			Venetian Awning Drop		
		0.65	0.60	0.55	0.65	0.60	0.55
South	7 AM 5	..	..	..	..	..	..
	8 ↓ 4 ↑	4	8	13	7	14	22
	9 ↓ 3	2	4	7	5	11	18
	10 2 PM	..	2	5	2	5	7
East ↓ West ↑	7 AM 5 ↑	7	13	22	7	13	22
	8 ↓ 4	..	7	15	..	7	15
	9 ↓ 3 PM	..	..	..	..	..	..

NOTE: Based on 3 FT x 5 FT window, awning extension 5 in. & 10 in. for conventional and venetian type awnings respectively.

window glass (data in Table 2C are about 1 to 3 percent low for ordinary window glass).

It was found that the total heat gains given in Table 2C could also be used for heat absorbing glass in combination with awnings provided the sunlit portion of the glass is not large. The reduced transmitted gains through the heat absorbing glass were largely offset by the convection-radiation gains resulting from higher glass temperatures.

The total heat gain through a window shaded by an awning having less than a 70 percent drop may, therefore, be determined by adding the 2 components of the gain which may be calculated as follows:

1. Heat gain through sunlit glass in the lower section of the window is equal to the sunlit glass area multiplied by the appropriate value from Table 2D.

2. Heat gain through the remainder of the window is equal to the area of the remainder multiplied by the appropriate value from Table 2C.

Example:—A southerly oriented window having an ordinary window glass 3 ft wide x *5 ft long is shaded by a conventional type of dark green canvas awning positioned to the wall*

with an extended head-rod clamp. The drop of the awning covers 55 percent of the entire glass height. Calculate the total heat gain through the window at 3 p.m. for: (1) 75 *F indoor temperature and a normal dark foreground;* (2) 80 *F indoor temperature and a normal dark foreground;* (3) 80 *F indoor design temperature and a light color foreground.*

Solution:—As the drop of the awning is less than 70 percent of the total glass height, the amount of sunlit area at the lower section, if any, should be determined. From Table 3, the sunlit area on the lower section is about 7 percent of the total glass surface. Hence, the sunlit area = 3 × 5 × 0.07 = 1 sq ft (approx.) and the shaded area = 3 × 5 − 1 = 14 sq ft.

(1) Total heat gain for a 75 *F indoor temperature and normal dark foreground*: For the shaded area (from Table 2C, column 2): 14 × 36 = 504 Btu per hr; for the sunlit area (from Table 2D, column 2): 1 × 63 = 63 Btu per hr; total heat gain through the entire glass surface = 567 Btu per hr.

(2) Total heat gain for an 80 *F indoor temperature and normal dark foreground*: The air temperature difference for this example is 95 − 80 instead of the 95 − 75 differential on which Table 2 values for 3:00 p.m. is based. Allowing 1 Btu per (hr) (sq ft) correction per Fahrenheit degree difference in temperature differential, the reduction in the heat gain is 15 × 5 = 75 Btu per hr. Hence the total heat gain is 567 − 75 = 492 Btu per hr.

(3) Total heat gain for an 80 *F indoor design temperature and a light color foreground*: The additional heat gain due to the increase in the transmitted component of the heat gain, as taken from Table 2A, column 1, is 7 Btu per (hr) (sq ft). Hence the total heat gain is 492 + 7 × 15 = 597 Btu per hr.

DISCUSSION OF RESULTS

The mathematical analysis of the transmitted and convected-radiated components of the total heat gain through canvas-awning-shaded windows is in fairly good agreement with the experimental data, considering the difficulty experienced in securing desirable weather conditions, and in measuring the ground reflected and sky components of diffuse solar radiation and the low temperature radiation received from the surroundings.

Reflected solar radiation received by the underside of the awning and reflected from it makes some contribution to the transmitted energy depending upon the intensity of direct solar radiation reflected from the wall surface surrounding the window and the reflectance of underside surface of the awning. Re-reflections between glass and awning are negligible for materials having low reflectance. If the awning material itself transmits solar radiation, as in the case of the plastic awning, the solar energy transmitted through the glass due to this component can be a significant amount.

The convection-radiation component of heat gain, which is related to the amount of heat absorbed by the awning and glass, the outdoor temperature, and the venting characteristics of the awning, calculated for a combined outside convection-radiation conductance of 3 Btu per (hr) (sq ft) (F deg), correlates well with test data as shown in Fig. 5. The limitation to the general application of this relation to all kinds of awnings is the necessity of determining experimentally the constant C appearing in the abscissa of Fig. 5. This constant depends on the venting of the awning. The conventional type of canvas awning with the standard-head-rod clamp and the venetian type of awning may represent the 2 extremes in the venting of canvas awnings. Furthermore, at wind velocities above 5 to 6 mph, the warm air under the awning is carried away by the wind. For awnings having low solar absorptance on the outer surface, venting is not important.

To illustrate the order of magnitude of the solar heat excluded by an awning over a period of a day, the total heat gains through 100 square feet of regular plate glass with and without awnings, are compared in Table 4. These data were obtained from Tables 2C and 2D.

CONCLUSIONS

1. The performance of canvas awnings in reducing the total heat gain into a room varies with type, color, and venting of the awning, and with orientation and the time

TABLE 4—HEAT EXCLUSION BY AWNINGS[a]

ORIENTATION OF WINDOW	TYPE OF GLASS AND AWNING[b]	HEAT GAIN PER 100 SQ FT GLASS SURFACE, BTU/DAY	HEAT EXCLUDED BY THE AWNING	
			BTU/DAY	PERCENT
South[c]	Regular plate glass alone	62200	0	0
South	Glass with conventional type white canvas awning	22500	39700	64
South	Glass with conventional type dark green canvas awning	27700	34500	55
South	Glass with conventional type dark green plastic awning	35600	26600	43
West[d]	Regular plate glass alone	84200	0	0
West	Glass with conventional type white canvas awning	19500	64700	77
West	Glass with conventional type dark green canvas awning	23900	60300	72
West	Glass with conventional type dark green plastic awning	34800	49400	59

[a] Data are for a window facing a dark foreground, an awning having a 70 percent drop, and for a typical design day (Aug. 1) at 40 deg. North Latitude.
[b] Awnings mounted with extended-head-rod clamps.
[c] For period from 8 a.m. to 4 p.m.
[d] For period from 12 noon to 5 p.m.

of day. The transmitted and convected-radiated components and the total heat gain for canvas awnings in combination with glass for East, South, and West orientations and for various hours of the day can be calculated from the data presented in Tables 2A, B, C, D, and Table 3.

2. An awning having a high solar absorbing surface on the outside, absorbs much of the solar radiation, causing its temperature and the temperature of air under it to rise, which in turn increases the glass temperature and the convection-radiation gain. On the other hand, the temperature of an awning having a low solar absorbing surface on the outside (white) will remain about the same as the outdoor air temperature. Thus, air under the awning is not appreciably heated and venting of such an awning is not important.

3. For wind velocities above 5 to 6 mph, venting of the awning is not important.

4. A light foreground in the presence of direct sunlight may approximately double the transmitted gain for a canvas awning-glass combination. For a plastic awning, the increase is approximately equal to the increase in transmitted gain for a canvas awning.

5. The use of heat absorbing glass instead of regular plate glass in combination with

awnings causes only a few Btu per (hr) (sq ft) reduction in the total heat gain, provided the sunlit portion of the glass is not large.

6. Over that period of a day during which an awning prevents the direct sun from falling upon the glass, on a southern exposure, 55 to 65 percent of the heat gain through the window is excluded by a canvas awning, and on a western exposure, the saving is 72 to 77 percent.

ACKNOWLEDGMENT

The authors acknowledge with thanks the suggestions and guidance of the members of the Technical Advisory Committee on Heat Transfer through Fenestration and the assistance and helpful criticism of their colleagues of the ASHAE Research Laboratory Staff.

REFERENCES

1. ASHVE RESEARCH REPORT No. 1333—Measurement of Solar Heat Transmission through Flat Glass, by G. V. Parmelee, W. W. Aubele, and R. G. Huebscher (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 168).
2. ASHVE RESEARCH REPORT No. 1442—Radiant Energy Emission of Atmosphere and Ground, by G. V. Parmelee and W. W. Aubele (ASHVE TRANSACTIONS, Vol. 57, 1952, p. 85).
3. ASHVE RESEARCH REPORT No. 1399—Heat Flow through Unshaded Glass—Design Data for Use in Load Calculation, by G. V. Parmelee and W. W. Aubele (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 371).
4. Radiant Heating and Cooling, Part I (Angle Factors for Calculations on Radiant Heating and Cooling) by C. O. Mackey, L. T. Wright, R. E. Clark and N. R. Gay (*Cornell University Bulletin No.* 32).

APPENDIX A

SOLAR RADIATION FALLING UPON THE GLASS SURFACE UNDER THE AWNING

Consider an awning and glass assembly, in which the awning shades all of the glass surface so that no direct beam falls upon the glass, and the awning material is opaque to solar radiation. The solar radiation received by the glass surface is largely due to the following components:

1. Ground-reflected and sky-diffuse solar radiation entering through the openings of the awning and falling upon the glass surface.
2. Ground-reflected and sky-diffuse solar radiation entering through the openings, and after being reflected from the underside surface of the awning, falling onto the glass surface.
3. Solar radiation reflected from the wall surrounding the window onto the underside of the awning, and in turn from the awning onto the glass.

The amount of re-reflected radiation between the glass and awning was relatively small, and was considered negligible.

In order to evaluate the foregoing quantities, the glass surface was arbitrarily divided into 2 sections:

Section 1: Glass surface above the lower edge of the awning.
Section 2: Glass surface below Section 1.

The amount of solar radiation falling upon each section was calculated in the following manner:

(*a*) Solar radiation falling upon Section 1 of area (A_1) consisted of the following components:

1. Ground-reflected solar radiation entering through the lower opening of area (A_3), $I_{GV}A_3F^{*}_{3-1}$.

2. Diffuse radiation from the sky entering through the top opening of area (A_6), $I_{SV}A_6F_{6-1}$.

3. Diffuse solar radiation both from the sky and that reflected from the ground entering through the side opening of area (A_7), $(I_{GV} + I_{SV})\ A_7F_{7-1}$.

4. Diffuse solar radiation entering through the lower, top and side openings and falling upon the underside of the awning of area (A_4), from where it is reflected onto Section 1 of the glass, $[I_{GV}A_3F_{3-4} + I_{SV}A_6F_{6-4} + (I_{GV} + I_{SV})\ A_7F_{7-4}]\ R_4F_{4-1}$.

(*b*) Solar radiation falling upon Section 2 of area (A_2) consisted of the following components:

NOMENCLATURE

A = surface area, square feet.
C = constant used in determining convection-radiation heat gains (Fahrenheit) (hour) (square foot) per Btu.
CR_{gi} = convection-radiation heat gain from the glass into the room, Btu per (hour) (square foot).
E = Stefan-Boltzmann constant multiplied by the temperature of the surface in Fahrenheit degree absolute to the fourth power, Btu per (hour) (square foot).
F = shape factor, dimensionless.
h_c, h_r, h_{cr} = surface conductance for convection, radiation, and convection-radiation, Btu per (hour) (square foot) (Fahrenheit degree).
H = mixing coefficient, Btu per (hour) (Fahrenheit degree).
I_{dV}, I_{DV} = intensity of total diffuse and direct solar radiation, on a vertical surface, Btu per (hour) (square foot).
I_{SV}, I_{GV} = intensity of sky-diffuse and ground-reflected solar radiation on a vertical surface, Btu per (hour) (square foot).
I_{dH}, I_{DH} = intensity of total diffuse and direct solar radiation on a horizontal surface, Btu per (hour) (square foot).
I_{DN} = intensity of direct solar radiation normal to the sun beam, Btu per (hour) (square foot).
αI_a, αI_g = solar energy absorbed by the awning and glass, Btu per (hour) (square foot).
t = temperature, Fahrenheit.
t'_o = temperature of air under the awning, Fahrenheit.
R = reflectance, dimensionless.
T = transmittance, dimensionless.
β = altitude of sun above a horizontal plane, degrees.

Subscripts

i, o — refer to inside and outside.
a, g — refer to awning and glass.

1. Diffuse solar radiation from the sky and reflected solar radiation from the ground, $[I_{GV} + I_{SV}\ (1 - F_{2-3})]\ A_2$.

2. Diffuse solar radiation reflected from the underside of the awning of area (A_4), $[I_{GV}A_3F_{3-4} + I_{SV}A_6F_{6-4} + (I_{GV} + I_{SV})\ A_7F_{7-4}]\ R_4F_{4-2}$.

Furthermore, the underside of the awning received solar radiation reflected from the wall surface surrounding the window, which, in turn was reflected onto the entire glass surface. This was

$$(I_{DV} + I_{dV})\ R_{wall}\ A_4F_{4-wall}\ R_4F_{4-(1+2)}$$

The sum of the foregoing components of solar radiation represents the total diffuse solar radiation falling upon the entire glass surface. The solar energy transmitted

* Shape factors in Appendix A are all for a quarter sphere.

through per unit area of the glass surface was obtained by dividing this sum by the area of the glass surface and multiplying it by the transmittance of glass for diffuse solar radiation.

APPENDIX B

THE GROUND-REFLECTED COMPONENT OF SOLAR RADIATION

Intensity of the ground-reflected solar radiation falling upon a vertical surface depends largely upon the intensity of solar radiation on the foreground and the reflectance of the foreground. The intensity of solar radiation on a horizontal foreground irradiated by the sun is $(I_{dH} + I_{DN} \sin \beta)$.

From the pyrheliometer measurements of the ground reflected solar radiation falling upon the vertical surface, it was found that, for the usual dark surroundings in front of the calorimeter, (I_{GV}) correlated approximately with the following relation:

$$I_{GV} = 0.06\ (I_{dH} + I_{DN} \sin\beta)$$

This relation was then used in computing the ratio (I_{GV}/I_{dV}), for calculations of the transmitted component of the heat gain. The values of I_{dH}, I_{DN}, I_{dV}, and β were taken from THE GUIDE.

APPENDIX C

CONVECTION-RADIATION HEAT GAIN FROM THE GLASS INTO THE ROOM

Equating the rate of solar energy absorbed per unit area of glass surface to the convection-radiation losses from the same surface:

$$\alpha I_g = CR_{gi} + h_{co}\,(t_g - t'_o) + h_{ro}\,(t_g - t_a)\,F_{g-a} + h_{ro}\,(t_g - t_o)\,(1 - F_{g-a}) \quad . \quad . \quad \text{(C-1)}$$

where

CR_{gi} = the convection-radiation gain from the glass into the room.

Writing a similar heat balance equation for the awning:

$$\alpha I_a = h_{co}\,(t_a - t_o) + h_{co}\,(t_a - t'_o) + h_{ro}\,(t_a - t_o)\,F \quad . \quad . \quad . \quad \text{(C-2)}$$

where

$$F = 1 + F_{a-1} + F_{a-\text{opening}}$$

For the purpose of this equation, the temperatures of the glass and the surroundings were assumed to be equal to the temperature of the outside air, t_o.
Solving for t_a in Equation C-2 and substituting in Equation C-1,

$$\left(\frac{1}{h_{cro}}\right)(CR_{gi}) + t_g = \frac{1}{h_{cro}}\,\alpha I_g + \left[\frac{h_{ro}F_{g-a}}{h_{cro}(2h_{co}+Fh_{ro})}\right]\alpha I_a$$
$$+ t_o\left(\frac{h_{ro}}{h_{cro}}\right)\left[1 - \frac{h_{co}F_{g-a}}{2h_{co}+Fh_{ro}}\right]$$
$$+ t'_o\left(\frac{h_{co}}{h_{cro}}\right)\left[1 + \frac{h_{ro}F_{g-a}}{2h_{co}+Fh_{ro}}\right] \cdots\cdots \text{(C-3)}$$

where, $h_{cro} = h_{co} + h_{ro}$

Taking the combined convection-radiation conductance*, h_{cro} as 3 Btu per (hr) (sq ft) (F deg) and the radiation conductance‡, h_{ro} as 1.2 Btu per (hr) (sq ft) (F deg), and substituting the numerical values of the shape factors, Equation C-3 may be reduced to the following form for conventional type awnings:

$$1/3\ CR_{gi} + t_g = (0.05\ \alpha I_a + 1/3\ \alpha I_g) + 1/3\ t_o + 2/3\ t'_o \quad . \quad . \quad \text{(C-3a)}$$

In this equation, other conditions being the same, t'_o is a function of the rate of removal of the hot air collected under the awning. Assuming that the air under the awning receives heat by convection from the awning only, and that this heat is carried away by air circulation only, the heat balance Equation C-4 can be written:

$$A_a\, h_{co}\, (t_a - t'_o) = H\, (t'_o - t_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(C-4)}$$

where H is determined from the test data. t'_o was solved by eliminating t_a from Equations C-2 and C-4. Substituting the value of t'_o thus obtained in Equation C-3a and putting in the respective values of the constants, Equation C-3a becomes as follows:

$$1/3\ CR_{gi} + t_g = (C\,\alpha I_a + 1/3\ \alpha I_g) + t_o \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(C-3b)}$$

The value of C was found to vary with the type of awning, the type of head-rod clamp, and the wind velocity. Values determined from experimental results, are as follows: For conventional awnings and wind velocities up to approximately 5 mph, C equals 0.07 and 0.10 for extended and standard type head-rod clamps respectively. For wind velocities above 5 mph, $C = 0.05$ for either type of head-rod clamp. For venetian type of awnings, a value of 0.05 may be used for any wind velocity and any type of head-rod clamp. Although weather conditions permitted the collection of only a few data at wind velocities below 2 mph, it is believed that the values may be applied without serious error for such conditions.

From Reference 3,

$$CR_{gi} = 0.27\ (t_g - t_i)^{0.25}\ (t_g - t_i) + 0.938\ (E_g - E_i) \quad . \quad . \quad . \quad \text{(C-5)}$$

From Equations C-3b and C-5, the convection-radiation gains were calculated for an inside design temperature of 75F, and were plotted in Fig. 5 as a function of the right hand term of Equation C-3b.

DISCUSSION

D. J. VILD, Toledo, Ohio (WRITTEN): The authors have succeeded in presenting in an easily understandable form a paper dealing with complex heat-transfer relationships. Data contained in this paper have been needed by residential air-conditioning engineers for many years and every attempt should be made to interpret it for inclusion in the ASHAE GUIDE as soon as practical.

It is interesting to note the heat gain due to transmitted and absorbed solar energy is virtually constant for wide ranges of solar azimuth and altitude for the various canvas awnings investigated. Solar azimuth has no significant effect between 0 and 60 degrees and solar altitude has little effect in cases where the shadow line reaches the bottom of the glass. Also with the use of extended head rod clamps the heat gains are essentially equal with either conventional or Venetian type awnings. These results are not altogether startling and are mentioned as items to be considered in preparing the data for THE GUIDE. Heat gains through awnings cannot be interpreted in the form of shade factors, such as are used for louvered shading devices, and other means as just mentioned

* Calculated values based on $h_{cro} = 3$ agreed well with experimental values.

‡ For the temperature ranges encountered during the tests for the glass and awning, and taking into consideration the emissivities of these surfaces, an average value 1.2 Btu per (hr) (sq ft) (F deg) was taken for the radiation conductance.

must be used. Supplementary information regarding the profile angles necessary to determine the sunlit glass areas would be needed for various latitudes, time of year, and orientation. With the addition of the heat transfer due to air-to-air temperature difference the total heat gain may be determined f-. all seasons. The importance of calculating air-conditioning loads for all times of the year should be stressed. The practice of designing air conditioning for an August 1 design day without a check of fall and winter cooling loads frequently results in the choice of undersized equipment.

A question arises regarding the determination of the values for the abscissa of Fig. 5. How is the quantity αI_a determined in a practical application? With variously oriented surfaces of the awning and hence variations in incident solar radiation, is this not an extremely difficult value to calculate? Also, how effective are the various awnings in reducing the heat gain on exposures oriented away from the sun and how effective are they in reducing the heating load?

W. P. CHAPMAN, Milwaukee, Wisc., (WRITTEN): The concise logical presentation of this paper makes it easy to understand and to apply. I believe, therefore, that the authors, in their attempt to simplify the presentation as much as possible, certainly have anticipated my comment. At any rate I would like them to comment on the following suggestion.

The information presented in Table 4 of this paper indicates that the effectiveness of awnings might possibly be explained by the following equation:

$$Q_a = K_a Q_w$$

where

Q_a = heat gain through the window with an awning, Btu per (hr) (sq ft)
K_a = awning shading factor, dimensionless
Q_w = heat gain through a bare window (that is, without awning) Btu per (hr) (sq ft).

If such an equation is feasible then I would suggest that the values for K_a be given in a table arranged in columns and rows. The rows could be the values of the solar altitude going from 0–90 degrees and the columns could be for wall-solar azimuth going from 0–90 degrees. The values of K_a would lie between zero and one. The azimuth would depend upon the wall orientation and the time of day; whereas, the altitude would be a function of the date and geographical location of the site.

Accompanying such a table could be a set of charts similar to those published by Irving F. Hand of the U. S. Weather Bureau in *Air Conditioning, Heating and Ventilating*, October 1948. Mr. Hand's charts enable the reader to determine the solar altitude and azimuth for any minute of the day for any point in the United States.

I would think that the advantage of such a table and the equation as mentioned would be that all of the data previously given for various types of glass could be modified to allow for the particular awning to be used. This would require a separate table for each awning, but in this case the tables are small and only 5 would be needed to present the data given in this paper.

AUTHORS' CLOSURE (Mr. Ozisik): We are grateful for the comments on this paper.

Mr. Vild pointed out in his discussion that the term, αI_a, which is the solar energy absorbed by the awning, is difficult to evaluate and we agree. The reason that the equation of solar energy absorbed was not included is that it is too complex to be of practical use. However, for the solar design conditions given in THE GUIDE and for a solar absorptance of unity on the outer surface of the awning, αI_a can be expressed as a function of the total incident radiation falling on the vertical wall having the same orientation as the window and the incidence angle of the solar beam on the wall. This relation is as follows:

INCIDENCE ANGLE	$\alpha I_a/(I_{DV} + I_{dv})$
0	0.52
20	0.66
40	0.90
60	1.22
80	2.15

and for a given awning, these values should be multiplied by the solar absorptance of the awning.

The effectiveness of awnings in reducing the heat gain on exposures oriented away from the sun is about 65 percent of that transmitted through the glass alone.

In regard to the effect of awnings in reducing the heat loss in winter time, we do not have any data on heat losses with canvas awnings. However, we have run tests on roller shades for both winter and summer conditions; and believe that the curve of Fig. 5 for convection-radiation heat gain can be extended downward without a serious error in determining the heat losses under winter conditions.

Mr. Chapman pointed out a simple method for presenting complex data as a function of the total gain through the glass alone and a constant varying with the solar altitude and solar azimuth. We shall consider this excellent suggestion to simplify future presentations.

1646

No. 1646

ACTIVATED CHARCOAL FOR AIR PURIFICATION

By H. L. BARNEBEY*, COLUMBUS, OHIO

ACTIVATED charcoal adsorption is a positive method for the control of odors and other unwanted vapors. It can be used in almost every situation where contaminants in vapor form should be removed from an enclosed space. To be effective, the system must be correctly designed and properly maintained.

The object of this paper is to give methods for determining the quantity of activated charcoal required per year of operation and the best type of adsorption system. Some of the basic approaches to the design of activated charcoal purification systems, the conversion factors, and tables of indices are presented here for the first time. Certain of the figures are based on a limited amount of numerical data. It is expected that the methods will be improved and inconsistencies ironed-out as their use increases.

Use of activated charcoal in connection with heating, cooling, and ventilating (Fig. 1) permits one or more of the following results.

1. Increased well-being and efficiency of personnel by removing stuffiness and irritating vapors.
2. Recirculation of all or part of the ventilating air, saving heat in winter and refrigeration in summer.
3. Purification of outside air used for ventilation or pressuring.
4. Ventilation of spaces not conveniently connected to central ventilating, heating, or cooling systems, *i.e.*, toilets.
5. Use of one circulating system for all rooms no matter what odors are released in the individual areas.
6. Elimination of odors from air exhausted to the atmosphere.
7. Recovery of values from vapors present in the space.
8. Increased safety by removing combustible or toxic vapors.
9. Elimination of toxic gases resulting from military or industrial emergencies.

Activated charcoal increases air purity in two ways:

1. Removing impurities by adsorption;
2. Permitting recirculation of air—outside air is not brought in and therefore atmospheric impurities are left outside.

---

* Vice President, Barnebey-Cheney Co.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

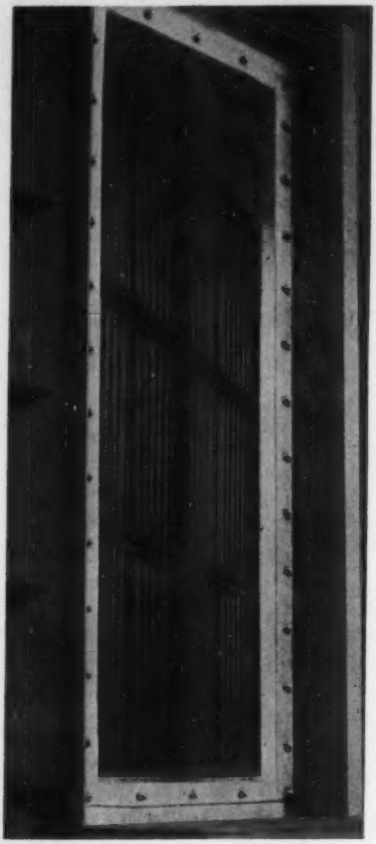


FIG. 1—BANK OF ACTIVATED CHARCOAL FILTERS

In recirculation systems, air is blown through a bed of activated charcoal or past surfaces of activated charcoal. If there is a high rate of air-turnover for uniformity or heat transfer purposes and a relatively low rate of odor release, only part of the air need pass through an activated charcoal bed or the entire flow can go through a partial by-pass type of filter and obtain the advantages of low resistance and low first cost.

Outside air is not pure in most industrial and urban locations and may not be suitable for ventilation purposes. In the case of an atmospheric or military emergency, ventilation with unpurified outside air presents exceptional hazards. For purifying outside air, exhaust air, or in recirculation systems with high odor level,

the activated charcoal filter should have high purification efficiency for each passage of air.

NATURE OF ODORS

The word *odor* is used herein to denote all of the types of contaminants, irritants, or noxious gases carried by air in vapor form which may be removed by purification with activated charcoal. An odor is a chemical compound or mixture of compounds in vapor form which affects the nose in such a way as to create the sensation of smell. Most of these substances have odors even though the effect on the nose may not be as serious as the irritation to other parts of the body. Even below the concentrations which can be recognized as odors, many of these vapors may cause stuffiness or drowsiness.

TABLE 1—SENSORY ODOR INTENSITY SCALE

DEGREE OF ODOR INTENSITY	DESCRIPTION
0	Odorless
1	Threshold
2	Definite
3	Strong
4	Overpowering

It requires only a very small amount (1 ppm, for example) of the vapor of certain chemical compounds in the air to give the sensation of odor. The quantity that is just barely perceptible as odor is called the threshold point. Ordinarily it requires about 10 times that concentration to give a definite sensation of odor, and another 10-fold increase before the odor is considered strong. One of the useful sensory scales for evaluating odor intensities is given in Table 1. This allows numbers to be assigned to odor intensities and makes it possible to average the results from odor tests.

Many gases and vapors are poisonous and there is a limit to the concentrations which the human body can tolerate under long exposure. Tables of maximum allowable concentrations are found in THE GUIDE 1957 (Table 3, p. 158) and other ventilating manuals.

NATURE OF ACTIVATED CHARCOAL

Activated charcoal has an internal sub-microscopic sponge structure consisting of tiny capillary passages, not greatly larger than the size of the molecules that are adsorbed. This is created in the manufacture of the adsorbent by burning out part of the carbon substance to form the internal surface. The terms *Activated Charcoal* and *Activated Carbon* have almost the same meaning and for most purposes can be considered synonymous.

Activated charcoal removes gases and vapors by adsorption. It does not remove ordinary particulate matter such as dust or pollen, but takes up particles so fine that they act in almost the same manner as a gas or vapor. A typical purifi-

cation system employs mechanical filters or electrostatic precipitators to remove dust and other particulate matter and activated charcoal to remove unwanted gases and vapors.

The adsorptive capacity of activated charcoal for odors in most applications is proportional to the evaluation obtained using an accelerated test with a pure chemical substance approximating the properties of typical odors and in which the charcoal bed is operated to the break-through point. Such a test was also found to be a good measure for gas mask charcoals and has been used by the U. S. Government for many years. This is known as the Accelerated Chloropicrin Test. An activated coconut shell charcoal having a value of 50 min by this test has been

Fig. 2—Activated Coconut Shell Charcoal

widely used in air purification. More activated charcoal of lower activity is required to do a given job and conversely a smaller amount of higher activity carbon is needed. For example, to do the work of 50-minute charcoal, there would be required at least a 25 percent greater quantity of 40-min charcoal.

Activated coconut shell charcoal (Fig. 2) has its adsorptive area in the form of very fine capillary pores which can remove and retain all types of vapor impurities even though they are present in very dilute form. Certain other types of activated carbon (such as prepared from coal or wood) have much larger pores and do not have the power of removing the lower molecular weight materials from dilute concentrations. As a specific example of this, tests have shown that of all commercial activated carbons available, coconut shell charcoal does the best job in critical odor control problems and in apple storages.

One pound of activated coconut shell charcoal has the capacity to remove from air all of the odor that a person would breathe in a normal lifetime. However, in purifying air for a living or working space, it is necessary to purify a great deal more air than is actually breathed by the occupants, perhaps 100 times more because the person has to have space to move about.

Quantity of Activated Charcoal Required

There are 3 main steps in applying activated charcoal to the control of odors in a specific application.

1. Determination of the amount and type of odor in terms of pounds of activated charcoal required per year.

2. Analysis of the cost of activated charcoal purification and the benefits to be received.

3. Settling the details of the application including air circulation rate, activated charcoal filter type, distribution of air flow, and installation details.

The amount of activated charcoal needed to purify a given space depends on the types of contaminants, the amount to be removed (this depends on the concentration, the proportion of the time that the concentration is present, the uniformity of vapors in the space, and related factors), and the type of activated charcoal used.

TABLE 2—CAPACITY OF 50-MIN ACTIVATED COCONUT SHELL CHARCOAL FOR VAPORS (CONDENSED, SEE APPENDIX TABLE A-1 FOR FULL TABLE)

2–Acetaldehyde	1–Carbon Dioxide	3–Hydrogen Sulfide
4–Acetic Acid	1–Carbon Monoxide	4–Isopropyl Alcohol
3–Acetone	4–Carbon Tetrachloride	4–Masking Agents
3–Acrolein	3–Chlorine	4–Mercaptans
4–Alcohol	4–Chloropicrin	4–Ozone
2–Amines	4–Cigarette Smoke	4–Perfumes, cosmetics
2–Ammonia	4–Cresol	4–Perspiration
3–Animal Odors	3–Diesel Fumes	4–Phenol
3–Anesthetics	4–Disinfectants	2–Propane
4–Benzene	4–Ethyl Acetate	4–Pyridine
4–Body Odors	1–Ethylene	4–Ripening Fruits
2–Butane	4–Essential Oils	4–Smog
4–Butyl Alcohol	2–Formaldehyde	3–Solvents
4–Butyric Acid	4–Gasoline	4–Stuffiness
4–Cancer Odor	4–Hospital Odors	4–Toluene
4–Caprylic Acid	4–Household Smells	4–Turpentine

The capacity index has the following meaning:
4–High capacity for all materials in this category. One pound takes up about 20 percent to 50 percent of its own weight—average about ⅓ (33-⅓%). This category includes most of the odor-causing substances.
3–Satisfactory capacity for all items in this category. These constitute good applications but the capacity is not as high as for Category 4. Adsorbs about 10 to 25 percent of its weight—average about 1/6 (16.7%).
2–Includes substances which are not highly adsorbed but which might be taken up sufficiently to give good service under the particular conditions of operation. These require individual checking.
1–Adsorption capacity is low for these materials. Activated charcoal cannot be satisfactorily used to re move them under ordinary circumstances.

Each odor control situation requires a certain activated charcoal adsorption capacity per year to satisfactorily do the job. The required quantity of adsorbent can be supplied in the form of permanent-type-factory-reactivated cells, or the disposable type. The air might pass through a small-area thick charcoal filter or a larger area having a thin carbon bed. The amount of activated charcoal required per year might be supplied, for example, in the form of a filter containing twice the yearly requirement which is reactivated every second year, or in throw-away filters having a six month's life and containing half the yearly requirement.

In designing an adsorption system, the calculation of the amount of charcoal should be based on removing 100 percent of the odor. It takes a 90 percent removal to reduce the concentration from the category of *definite* to *threshold* and a 99 percent reduction to go from *strong* to *threshold*. After making an effective or noticeable odor reduction, it takes very little additional activated charcoal to remove the remaining odor.

The most difficult problem in applying any type of air purification is a determination of the quantity and type of contaminants that must be removed. With this information at hand the quantity of activated charcoal can be determined easily. Often a shortcut can be taken which combines the two into one step. Listed here are 7 methods. One or more of these is applicable to every situation.

It is best to figure the amount of charcoal required by several methods if applicable and compare the results.

A. Knowledge or analytical determination of specific odor-causing compounds and their amounts.
B. Inventory of individual odor sources.
C. Type of occupancy and volume to be purified.
D. Dilution by purified air using the tables in THE GUIDE of ASHAE.
E. Purification test using a small adsorber.
F. Survey based on smell.
G. Rules of thumb and experience.

TABLE 3—POUNDS OF 50-MIN ACTIVATED COCONUT SHELL CHARCOAL REQUIRED PER YEAR[a]

Residences	
Total per person	
Non-smoking	1
Smoking	2
Individual rooms per person	
Living room	0.5
Dining room	0.25
Kitchen	0.25
Toilet and bath	0.25
Bedroom	0.15
Laundry	0.15
Office	
Total per person	
Non-smoking	1
Smoking	2
Hotel, per person, average occupancy	2
School, per pupil	1.5
Hospital (room or ward), per bed	3
Laboratory (average), per person	3
Bar or tavern, per occupant	5

[a] Add up the amount of activated charcoal needed by units as indicated in the table. For example, if there are 6 people in an office, 3 of whom are smokers, and 3 non-smokers, a total of 9 lb for one year's service under average conditions is indicated.

Method A—Knowledge or analytical determination of specific odor-causing compounds and their amounts: This calls for knowledge of the operations going on within the area or an analysis of the contaminants in the air. If the odors are from a manufacturing process or similar operation, it may be possible to establish the types of vapors given off, their concentration, and the quantity which must be removed per year. In other cases this information can be obtained by analytical procedures. A method useful where the odor concentration is rather high involves freezing out the vapors at low temperature and subsequent chemical analysis.

Table 2 gives the relative capacity of 50-min activated coconut shell charcoal for most of the common air contaminants and many others that are found in specific applications. The figures are based on the removal of the substance from dilute concentration in air as is encountered in air recovery and air purification projects.

If the odor under consideration is listed in classification 4, then it can be assumed that each pound of activated charcoal will take up one-third of a pound of odor. Multiply the weight of odor to be removed by 3 to determine the quantity

of activated charcoal needed to do the job. If the compound is in class 3, multiply the weight of odor by 6.

Method B—Inventory of individual odor sources: An odor inventory (directly in terms of activated charcoal required) can be prepared using Table 3 or similar information for the situation under consideration.

Method C—Type of occupancy and volume to be purified: Given in Table 4 are the amounts of space (cubic feet) of some selected types which can be purified by one pound of activated charcoal per year.

In connection with many chemical processes, coating operations, and other locations where large amounts of volatile compounds are given off, it is possible to regenerate the activated charcoal in-place (Fig. 3) by steaming or other suitable treatment. In a typical situation, the activated charcoal might be desorbed and reused 5,000 times in one year which means that one pound could perform the same service as many pounds if in-place regeneration were not used. In the usual type

TABLE 4—CUBIC FEET OF SPACE WHICH CAN BE PURIFIED BY 1 LB OF ACTIVATED CHARCOAL PER YEAR[a]

Chemical processes	1/10 to 100
Chemical plants	50 to 500
Laboratories	50 to 500
Printing plants	10 to 500
Toilets	100 to 1000
Cold storage	100 to 1000
Offices	100 to 5000
Public buildings	200 to 1000
Homes	1000 to 5000

[a] This table is for general guidance only and gives the range in which the stated applications ordinarily fall. Although there is a variation of about 50 to 1 in typical air-conditioning situations, it is less than the difference between threshold and strong odor levels.

of air-conditioning applications, it is not feasible or necessary to use in-place regeneration.

The relative odor levels in the various types of locations where activated charcoal air purification can be used are listed in Table 5 as A, B, C, or D. Locations A contain the lowest amount of odor and D the highest. B and C represent levels in between. Many of the classifications are rather general so it was necessary to pick a typical average condition. The odor index for a specific situation could vary somewhat from that given in the table if special circumstances apply. In typical cases, one pound of 50-min activated coconut shell charcoal will purify the following cu ft of space for one year: A—2,000; B—800; C—300; and D—100.

For certain specific types of applications there may be rather accurate experience which dictates the amount of activated charcoal to use and the best method of applying it. For example, in cold storage rooms for apples, it has been shown that 6 lb of activated coconut shell charcoal and 100 cfm of recirculated air should be used during each storage period for each 1,000 bushels of apples.

Method D—Dilution by purified air using the tables in THE GUIDE *of ASHAE:* This is based on dilution by purified air. Table 1 (p. 113) and Table 3 (p. 297) of THE GUIDE 1957 show the amount of outside air which should be used for ventilation (elimination of odors and stuffiness). Activated charcoal is used to purify

TABLE 5—ODOR INDEX FOR TYPE OF SPACE (CONDENSED, COMPLETE TABLE A-2 IN APPENDIX)

C–Aircraft	B–Department stores	B–Offices
D–Air raid shelters	C–Drug stores	C–Photo dark rooms
D–Animal rooms	D–Funeral homes	D–Pollution control
A–Apartment buildings	A–Homes	C–Public toilets
C–Apple storage	C–Hospitals	C–Recreation rooms
B–Auditoriums	B–Hotels	B–Restaurants
C–Bars	C–Kitchens	C–Schools
C–Beauty shops	C–Locker rooms	B–Super markets
A–Churches	D–Meat packing plants	C–Telephone exchanges
C–Conference rooms	C–Morgues	B–Theaters

an equivalent amount of recirculated air and this is used instead of outside air. A standard practice for ventilating might indicate in a specific case that 30 percent of the recirculated air should be exhausted and replaced with fresh air. If this same amount is purified by activated charcoal rather than replaced, the result will be essentially the same. This means that of the recirculated air stream, 30 percent might be 95 percent purified by activated charcoal and mixed back in with the rest of the stream. This means that the recirculated stream has been purified to an extent of about 30 percent and the total stream is certainly not equivalent to 100 percent fresh air. But if the amount purified by activated charcoal is the same in quantity as the amount of fresh air, that might have been used, the result will be the same. If improved air quality is desired, a greater proportion of the recirculated stream should be purified.

Method E—Purification test using a small adsorber: A test can be made with a small activated charcoal purification unit (Fig. 4) to duplicate with a portion of the air or part of the space the type of treatment that can be expected with commercial activated charcoal filters. This test requires only a small amount of equipment and is the simplest approach to complicated odor control problems.

For most industrial and commercial applications, it is suggested that the test be run with a test unit which has an air capacity of 65 cfm and contains 3 lb of ac-

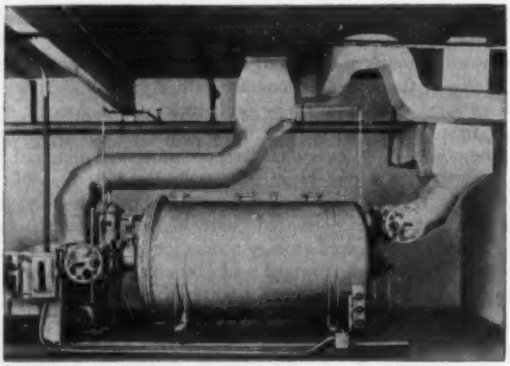


FIG. 3—ACTIVATED CHARCOAL UNIT WITH IN-PLACE REGENERATION

tivated coconut shell charcoal. This is a ratio of about 45 lb per 1,000 cfm which is that contained in commercial activated charcoal cells. The test unit should be mounted in the area under consideration and operated so that the air being treated by it will represent that to be handled in the commercial installation. By smelling the air discharged from the test unit, it can be determined whether the odors are being satisfactorily removed. The human nose sometimes becomes accustomed to odors after long periods of exposure and it is suggested that the person doing the testing should allow the discharge to blow on his face for a period of some minutes and after this exposure to deodorized air, then smell the surrounding area which has not been so purified. In some cases it may be convenient to isolate a small

FIG. 4—CANISTER CIRCULATOR UNIT

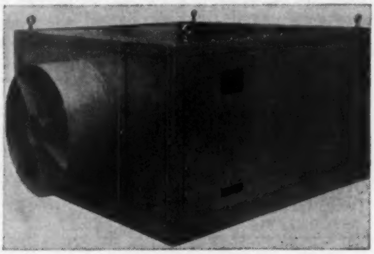


FIG. 5—CABINET TYPE PURIFYING UNIT

portion of the space, say 500 to 1,000 cu ft, and purify that part with the small purification unit.

This type of test cannot be done rapidly. The unit should operate for a long enough time to definitely show that the activated charcoal will have good life under the existing conditions and that it will not be necessary to reactivate so often that the application is not economically feasible. For example, if odors break through after one month's operation, it indicates that it will be necessary to reactivate the commercial installation after each month. This may be too often and entail too much expense. On the other hand, if the odor level is very heavy and there is much to be gained by completely purifying the air, it still may be worthwhile to do. For heavy duty applications where the test shows short life, the use of filters with thick activated charcoal beds may be indicated. Generally speaking, in most commercial and industrial situations, the adsorber cells have to be reactivated every 6 months to 2 years, depending on odor level. If no odor breaks through the test cell in 6 months, it can be assumed that the commercial equipment will function for at least 6 months without requiring reactivation.

It is possible in some cases to accelerate the test. If, for example, the ventilating system normally operates only 8 hr per day and 5 days per week, but the odor is

continuously present, then it may be possible to run the test cell 24 hr a day and 7 days per week. That means that one month's operation would be equivalent to 4.2 months of actual service. Also, it may be that the odor is more concentrated in a given area. If the test unit is run at such a location, it can be assumed that the saturation of the test cell will be accelerated. If the odor breaks through in one month, one can say, just as a guess, that this might represent anywhere from 5 months to a year in actual service on air having the average concentration of odors. It is somewhat dangerous to try to make accurate predictions by such a procedure because a slight difference in odor perception represents a tremendous increase in odor concentration.

If the test unit is operated for one or two months under average conditions it may then be possible to determine analytically the degree to which the activated charcoal's capacity has been depleted and give an estimate on the adequacy of charcoal treatment for the purpose.

TABLE 6—CORRELATION BETWEEN ODOR INTENSITY, VAPOR CONCENTRATION, AND SPACE WHICH CAN BE PURIFIED BY ONE POUND OF ACTIVATED CHARCOAL PER YEAR

DEGREE OF ODOR	CU FT PER LB OF CHARCOAL PER YEAR	LB OF ODOR PER MILLION CU FT	ODOR INDEX
Too little to smell	10,000	0.001	0
Threshold (just perceptible)	1,000	0.01	1
Definite	100	0.1	2
Strong	10	1.0	3
Overpowering	1	10	4
Toxic (process only)	1/10	100	5

It is sometimes possible to desorb the odor from the charcoal and determine the main types of chemical compounds involved, that is, whether they are organic compounds or inorganic materials and whether the major constituent is, for example, an alcohol, an ester, or an aldehyde. An exact analysis of the quantities of different individual odor-causing constituents is ordinarily difficult. There are many different chemical compounds which make up the average odor mixture and many of these are changed in composition upon being removed from the charcoal.

Method F—Survey based on smell: Table 6 is a rough correlation between odor intensity, vapor concentration, and space which can be purified by one pound of activated charcoal per year. With approximately 500,000 (more exactly, 525,600) min in a year and at a 5-min air change, there would be 100,000 air changes per year. Assume that activated charcoal refers to 50-min coconut shell charcoal; assume the odor is at the stated level one-third of the time, or 8 hr per day; assume that 1 lb of activated charcoal takes up one-third pound of odor. The odor level required to give a certain physiological reaction is a sort of typical average figure. Odor perception varies from person to person and also depends on temperature, humidity, and air velocity. Some odors are stronger than others and the amount required might be greater or less by a factor of 10 or more.

A 1,000 cubic foot space at the threshold level will require the treatment of 100 million cu ft of air per year at a 5-min change. If there is one-hundredth of a pound

of odor per one million cubic feet, this means that 1 lb of odor will be removed per year from this space, providing the stated level persists 100 percent of the time. Since it was assumed that it will persist one-third of the time, this gives 1/8-lb of odor. Also, since it has been assumed that 50-min activated charcoal will take up 1/8-lb of odor per lb, 1 lb of activated charcoal will do the job. This manner of calculating is approximate but illustrates the principles involved in odor control.

Method G—Rules of thumb and experience: A person experienced in odor control problems and in the use of activated charcoal can ordinarily make a reasonable selection of equipment based on rather meager information. The more information that is available, the more accurate can be the selection.

It is possible to make a rough determination of odor concentration by smell. An expert in odor control may be able to recommend the type of purification equipment to use based on an examination of the situation using his sense of smell. Whenever possible, this should be checked by one of the other methods.

Wide experience in odor control permits an expert to make recommendations based on knowledge of the performance of successful installations. He knows what type of equipment gave satisfactory results for similar problems (Fig. 5).

Types of Activated Charcoal Purification Systems

The type of activated charcoal filters to use and the amount of air circulation depend on the amount of charcoal required, how the odor is released in the room. the type of occupancy, the arrangement of the ventilation system, and similar factors. The higher the rate of odor release, the faster should be the air turnover and the more activated charcoal is required per year.

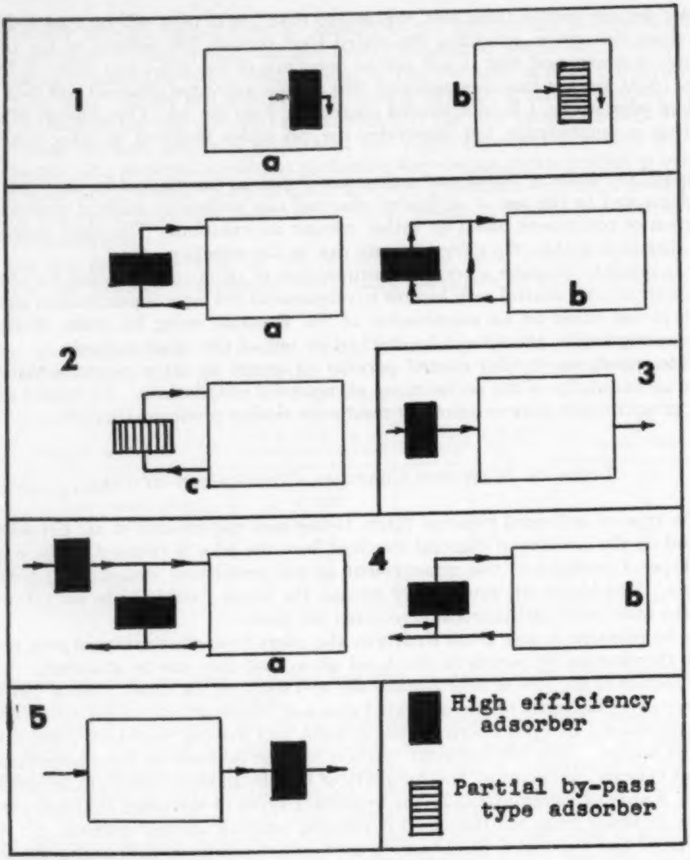
To be effective in any given situation, the odors from the ventilated area must reach the charcoal by means of circulated air so that they can be adsorbed. This is a function of the rate of odor release, size and shape of the room, type of ventilating system, placement of the activated charcoal filters or circulating units, eddy currents caused by open doors, partition walls, and moving machinery, and other related factors. The physiological reaction to odor depends on its concentration, the air velocity, temperature and humidity of the air, and the individual occupants.

Fig. 6 shows diagrammatically the important types of activated charcoal purification methods using self-contained circulating units or ducted systems.

Types 3 and 4 of Fig. 6 should not be used unless outside air is desirable for temperature, humidity, oxygen, or carbon dioxide control. Where there is some doubt about the oxygen sufficiency, bring in a small amount of purified outside air. It is easy to determine the oxygen content in the building under consideration or in a similar building. Such a test employs standard equipment and its use can be learned quickly.

Another approach where regulations or personal opinions have dictated a certain amount of outside air is to design the system so that ventilating air can be brought in from the outside if desired or shut off so that complete recirculation is employed. After the system is in operation, a few tests will demonstrate where complete circulation can be used to advantage.

As indicated in Type 5 of Fig. 6, activated charcoal can be used to advantage to purify air exhausted from kitchens, toilets, process operations, or other areas having high concentrations of contaminants. The air, after purification, can be discharged to the atmosphere without fear of pollution or can, in many cases, be



1. Self-contained unit
   (a) complete purification in each passage
   (b) partial purification in each passage
2. Recirculation system
   (a) complete purification per pass handling all the air
   (b) complete purification per pass handling part of the air
   (c) partial air by-pass filters handling all the air
3. Outside air (handling it all through activated charcoal)
4. Combination outside and recirculated air
   (a) through separate banks of charcoal filters
   (b) through one bank of activated charcoal filters
5. Exhaust system (alone or combined with one of the others)

FIG. 6—SYSTEMS OF APPLYING ACTIVATED CHARCOAL

reused in the ventilation of the building. Where the concentrations are high, thicker beds of activated charcoal should be used or in some cases the cells can be regenerated in-place. It is sometimes possible to recover useable and valuable material from exhaust air (Fig. 7).

Purification of all of the recirculated air by activated charcoal is the most foolproof way in many cases. There is little that can go wrong with such a system and it will operate effectively even though there are periodic overloads of odor. This may, however, be more expensive in the initial installation than a smaller but adequate system. An oversize system does not cost any more as far as operation

FIG. 7—ACTIVATED CHARCOAL INSTALLATION FOR RECOVERY OF VALUES IN GASES

is concerned. If extra activated charcoal is put in initially, it means that the cells will not have to be reactivated as often.

If there is doubt about the amount of odor to be removed or the rate of air change which should be employed, it is feasible to install a minimum bank of adsorbers and provide in the building plan for the addition of more adsorbers if they should be needed.

In cases of high odor concentration, it may be feasible to lighten the load on the activated charcoal adsorbers by using some type of pretreatment. The adsorbers should usually be protected with dust filters to prevent them from becoming mechanically loaded with particulate matter which interferes with the air flow. In the case of ammonia (for which activated charcoal has limited capacity), scrubbing with water either before or after the activated charcoal cells may be desirable. In the case of kitchen exhaust, adequate grease filters should be used ahead of the adsorbers. With heavy concentrations of smoke (which does not perform like a true vapor), it may be advantageous to use electrostatic precipitators ahead of the carbon filters.

Let us consider a charcoal filter having a ½ in. thickness of 6 x 10 mesh, 50-min activated coconut shell charcoal and sufficient in area so that the face velocity is

about 40 fpm. The air after passing through such a filter is generally odor-free and the equivalent of fresh air. Manufacturers rate the efficiencies of such adsorbers at approximately 95 percent. The efficiency might be higher than this, say 97 or 98, when the cell is first installed and may drop to about 90 just before the cell is reactivated or the activated charcoal changed. In most of the ventilating problems a 95 percent removal efficiency is sufficient to bring the odor level well below threshold and produce fresh air. Such a filter has a pressure drop of 0.2 in. of water.

An activated charcoal bed 1 in. thick rather than ½ in. thick might have an average efficiency during its useful life of about 98 percent. It might start out at 99 percent and drop to as low as 90 percent just before the charcoal is changed. Such a cell has a pressure drop of 0.30 to 0.35 in. of water at standard velocities.

TABLE 7—CORRELATION BETWEEN ODOR CONCENTRATION AND TYPE OF EQUIPMENT

Odor Index	Pounds of Odor per Million Cu Ft	Cubic Feet of Space per lb of Charcoal per Year	Activated Charcoal Equipment Type		
			Disposable Filters	Factory Reactivated Filters	In-Place Reactivated Units
0	0.001	10,000	X		
1	0.01	1,000	X	X	
2	0.1	100	X	X	
3	1.0	10		X	X
4	10	1			X
5	100	1/10			X

Everything else being equal, the service life (not initial efficiency) and resistance are proportional to the thickness of the activated charcoal bed. Commercial adsorber cells of the straight-through-factory-reactivated type are available with charcoal bed thicknesses from ½ to 4 in. and the efficiencies, depending on conditions of operation, can vary from 90 to practically 100 percent. If the cells are reactivated frequently the adsorption efficiency can be maintained at a high level. Deep carbon beds (frequently regenerated in-place) are used for very high concentrations such as are found in process applications. The partial by-pass types have efficiencies in the general range of 5 to 50 percent and the resistance to air flow is generally lower than the other types.

To purify 30 percent of the recirculated stream through a ½ in. thick charcoal filter of the type just described means that 30 percent of the air must be boosted to an additional pressure of 0.2 in. of water by a fan or the total air stream must be increased in pressure by this amount and 70 percent throttled through by-pass valves. Another way to solve the same problem at lower pressure drop is to use a partial by-pass type of filter having an efficiency of 30 percent and this might be done with a pressure drop of about 0.1 in. of water rather than 0.2 in.

A common error in building ventilation stems from a wrong assumption in the basic facts that apply to the problem, particularly with regard to the amount and type of odor to be removed. The usual result is an inaccurate prediction of how often the purification cells must be reactivated. Suppose, for example, a building has a volume of 300,000 cu ft. One engineer might decide that the air should be

changed every 15 min. This would mean a circulation of 20,000 cfm or would require 20—1,000 cfm adsorbers if all the recirculated air is purified. Another engineer might decide that for odor control or temperature control reasons, the rate of air change should be every 3 min. This would call for 100,000 cfm, or 100—1,000 cfm cells. The purification problem has not changed but there is a five-fold increase in the equipment required. The latter system with only 20 percent of the air passing through 20 cells might be equal to the first in odor control performance.

Table 7 gives a rough correlation between the odor concentration and the type of equipment recommended. As an example of the use of Table 7, consider a space

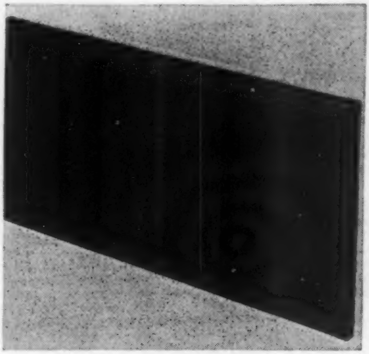


FIG. 8—DISPOSABLE ACTIVATED CHARCOAL FILTER

FIG. 9—1,000 CFM ADSORBER CONTAINS 45 POUNDS OF ACTIVATED CHARCOAL

of 40,000 cu ft with an odor index of 2, which means that there is a definite (but not strong) odor present. The table shows that one pound of activated charcoal per year is required for each 100 cu ft of space purified, or in this case, 400 lb.

By using standard commercial adsorbers (Fig. 9) which contain 45 lb of 50-min. activated coconut shell charcoal there would be needed the equivalent of 9 cells per year. By using 9 cells in parallel (9,000 cfm recirculation) the bank would be reactivated once each year. This would give an air change every 4½ min. If it is decided that a less rapid air change is satisfactory, say every 10 minutes, this would mean a circulation of 4,000 cfm and a bank of 4 cells can be used in parallel. To give the same purifying capacity would require that the bank be reactivated approximately once every 5 months.

In general, use the amount of activated charcoal required in the form of adsorber cells which best fit in with the ventilating, heating, or cooling system with regard to size, air velocity, and pressure drop. Complete detailed information can be obtained from the manufacturer's literature or by submitting the problem to specialists in activated charcoal air purification.

APPENDIX

TABLE A-1—CAPACITY OF 50-MIN ACTIVATED COCONUT CHARCOAL FOR VAPORS (EXTENSION OF TABLE 2 OF PAPER)

The capacity index has the following meaning:

4–High capacity for all materials in this category. One pound takes up about 20% to 50% of its own weight—average about $\frac{1}{3}$ (33-$\frac{1}{3}$%). This category includes most of the odor causing substances.

3–Satisfactory capacity for all items in this category. These constitute good applications but the capacity is not as high as for category 4. Adsorbs about 10 to 25% of its weight—average about $1/6$ (16.7%).

2–Includes substances which are not highly adsorbed but which might be taken up sufficiently to give good service under the particular conditions of operation. These require individual checking.

1–Adsorption capacity is low for these materials. Activated charcoal cannot be satisfactorily used to remove them under ordinary circumstances.

2–Acetaldehyde
4–Acetic Acid
4–Acetic Anhydride
3–Acetone
1–Acetylene
3–Acids
3–Acrolein
4–Acrylic Acid
4–Acrylonitrile
4–Adhesives
4–Air Wick
4–Alcohol
4–Alcoholic beverages
2–Amines
2–Ammonia
4–Amyl Acetate
4–Amyl Alcohol
4–Amyl Ether
3–Animal Odors
3–Anesthetics
4–Aniline
4–Antiseptics
4–Asphalt Fumes
3–Automobile Exhaust
3–Bacteria
4–Bathroom Smells
4–Benzene
3–Bleaching Solutions
4–Body Odors
4–Bromine
4–Burned Flesh
4–Burned Food
4–Burning fat
3–Butadiene
2–Butane
4–Butanone
4–Butyl Acetate
4–Butyl Alcohol
4–Butyl Cellosolve
4–Butyl Chloride
4–Butyl Ether
2–Butylene
2–Butyne
3–Butyraldehyde
4–Butyric Acid
4–Camphor
4–Cancer Odor
4–Caprylic Acid
4–Carbolic Acid
3–Carbon Bisulfide
1–Carbon Dioxide
1–Carbon Monoxide
4–Carbon Tetrachloride
4–Cellosolve
4–Cellosolve Acetate
4–Charred Materials
4–Cheese
3–Chemicals
3–Chlorine
4–Chlorobenzene
4–Chlorobutadiene
4–Chloroform
4–Chloro Nitropropane
4–Chloropicrin
4–Cigarette Smoke
4–Citrus and other fruits
4–Cleaning Compounds
3–Coal Smoke
3–Combustion Odors
4–Cooking Odors
3–Corrosive Gases
4–Creosote
4–Cresol
4–Crotonaldehyde
4–Cyclohexane
4–Cyclohexanol
4–Cyclohexanone
4–Cyclohexene
4–Dead Animals
4–Decane
4–Decaying Substances
4–Decomposition Odors
4–Deodorants
4–Detergents
4–Dibromoethane
4–Dichlorobenzene
3–Dichlorodifluoromethane
4–Dichloroethane
4–Dichloroethylene
4–Dichloroethyl Ether
3–Dichloromonofluormethane
4–Dichloro-Nitroethane
4–Dichloropropane
3–Dichlorotetrafluoroethane
3–Diesel Fumes
3–Diethyl Amine
4–Diethyl Ketone
4–Dimethylaniline
4–Dimethylsulfate
4–Dioxane
4–Dipropyl Ketone
4–Disinfectants
4–Embalming Odors
1–Ethane
3–Ether
4–Ethyl Acetate
4–Ethyl Acrylate
4–Ethyl Alcohol
3–Ethyl Amine
4–Ethyl Benzene
3–Ethyl Bromide
3–Ethyl Chloride
3–Ethyl Ether
3–Ethyl Formate
4–Ethyl Mercaptan
4–Ethyl Silicate
1–Ethylene
4–Ethylene Chlorhydrin
4–Ethylene Dichloride
3–Ethylene Oxide
4–Essential Oils
4–Eucalyptole
3–Exhaust Fumes
4–Female Odors
4–Fertilizer
3–Film Processing Odors
4–Fish Odors
4–Floral Scents
3–Fluorotrichloromethane
4–Food Aromas
2–Formaldehyde
3–Formic Acid
2–Fuel Gases
3–Fumes
4–Gangrene
4–Garlic
4–Gasoline
4–Heptane
4–Heptylene
3–Hexane
3–Hexylene
3–Hexyne
4–Hospital Odors
4–Household Smells
1–Hydrogen
2–Hydrogen Bromide
2–Hydrogen Chloride
3–Hydrogen Cyanide
2–Hydrogen Fluoride
3–Hydrogen Iodide
2–Hydrogen Selenide
3–Hydrogen Sulfide
4–Incense
4–Indole
3–Inorganic Chemicals
3–Incomplete Combustion
3–Industrial Wastes
4–Iodine
4–Iodoform
4–Irritants
4–Isophorone
3–Isoprene
4–Isopropyl Acetate
4–Isopropyl Alcohol
4–Isopropyl Ether
4–Kerosene
4–Kitchen Odors
4–Lactic Acid
4–Lingering Odors
4–Liquid Fuels
4–Liquor Odors
4–Lubricating Oils and greases
4–Lysol
4–Masking Agents
4–Medicinal Odors
4–Melons
4–Menthol
4–Mercaptans
4–Mesityl Oxide
1–Methane
3–Methyl Acetate
4–Methyl Acrylate
3–Methyl Alcohol
3–Methyl Bromide
4–Methyl Butyl Ketone
4–Methyl Cellosolve
4–Methyl Cellosolve Acetate
3–Methyl Chloride
4–Methyl Chloroform
3–Methyl Ether
4–Methyl Ethyl Ketone
3–Methyl Formate
4–Methyl Isobutyl Ketone
4–Methyl Mercaptan
3–Methylal

TABLE A-1 (Continued)

4–Methylcyclohexane	3–Pentylene	4–Smog
4–Methylcyclohexanol	3–Pentyne	4–Soaps
4–Methylcyclohexanone	4–Perchloroethylene	4–Smoke
4–Methylene Chloride	4–Perfumes, cosmetics	3–Solvents
3–Mildew	4–Perspirations	4–Sour Milks
4–Mixed Odors	4–Persistent Odors	4–Spilled Beverages
3–Mold	4–Pet Odors	4–Spoiled Food Stuffs
4–Monochlorobenzene	4–Phenol	4–Stale Odors
3–Monofluorotrichloromethane	3–Phosgene	4–Stoddard Solvent
4–Moth Balls	4–Pitch	4–Stuffiness
4–Naphtha (Coal tar)	4–Plastics	4–Styrene Monomer
4–Naphtha (Petroleum)	3–Poison Gases	4–Sulfur Compounds
4–Naphthalene	3–Pollen	2–Sulfur Dioxide
4–Nicotine	4–Popcorn and Candy	3–Sulfur Trioxide
3–Nitric Acid	4–Poultry Odors	4–Sulfuric Acid
4–Nitro Benzenes	2–Propane	4–Tar
4–Nitroethane	3–Propionaldehyde	3–Tarnishing Gases
2–Nitrogen Dioxide	4–Propionic Acid	4–Tetrachloroethane
4–Nitroglycerine	4–Propyl Acetate	4–Tetrachloroethylene
4–Nitromethane	4–Propyl Alcohol	4–Theatrical Makeup Odors
4–Nitropropane	4–Propyl Chloride	4–Tobacco Smoke
4–Nitrotoluene	4–Propyl Ether	4–Toilet Odors
4–Nonane	4–Propyl Mercaptan	4–Toluene
3–Noxious Gases	2–Propylene	4–Toluidine
4–Octalene	2–Propyne	4–Trichlorethylene
4–Octane	3–Putrefying Substances	4–Turpentine
4–Odors	4–Putrescine	4–Urea
4–Odorants	4–Pyridine	4–Uric Acid
4–Onions	2–Radiation Products	4–Valeric Acid
4–Organic Chemicals	4–Rancid Oils	4–Valericaldehyde
4–Ozone	3–Refrigerant-12	4–Vapors
4–Packing House Odors	4–Resins	4–Varnish Fumes
4–Paint and Redecorating Odors	4–Reodorants	4–Vinegar
4–Palmitic Acid	4–Ripening Fruits	3–Vinyl Chloride
4–Paper Deteriorations	4–Rubber	3–Viruses
4–Paradichlorbenzine	4–Sauerkraut	3–Volatile Materials
4–Paste and glue	4–Sewer Odors	4–Waste Products
3–Pentane	4–Skatole	3–Wood Alcohol
4–Pentanone	3–Slaughtering Odors	4–Xylene

Some of the contaminants listed in the table are specific chemical compounds, some represent classes of compounds, and others are mixtures and of variable composition. Activated charcoal's capacity for odors varies somewhat with the concentration in air, with humidity and temperature, and with the actual velocity used through the filters. The numbers given represent typical or average conditions and might vary in specific instances. The values in the table have been assembled from many sources including laboratory tests and field experience. In cases where numerical values were not available, the author has listed his opinion of the probable capacity based on general experience. The table should be used as a general guide only.

TABLE A-2—ODOR INDEX FOR TYPE OF SPACE (EXTENSION OF TABLE 5 OF PAPER)

Locations A contain the lowest amount of odor and D the highest. B and C represent levels in between. Many of the classifications are rather general so it was necessary to pick a typical or average condition. The odor index for a specific situation could vary somewhat from that given in the table if special circumstances apply. In typical cases, one pound of 50-min activated coconut shell charcoal will purify the following amount of space for one year: A = 2,000; B = 800; C = 300; and D = 100.

C–Adhesive manufacturing plants	D–Bank vaults	D–Chemical plants
C–Air conditioning systems	C–Banquet rooms	D–Chemical storage
C–Aircraft	C–Barber shops	D–Chlorine manufacture
B–Airline terminals	C–Bars	A–Churches
D–Air raid shelters	C–Basements	B–Circulating fans
B–Amusement places	B–Bathrooms	B–Circulating systems
D–Animal rooms	C–Beauty shops	C–Clinics
A–Apartment buildings	A–Bedrooms	B–Closets
A–Apartments	B–Binderies	C–Club houses
C–Apple storage	D–Biological processes	D–Coating processes
B–Art studios	D–Bomb shelters	C–Cocktail lounges
C–Athletic clubs	B–Book stacks	C–Cold storage plants
D–Atomic processes	C–Breweries	D–Collective protection shelters
B–Auditoriums	C–Buses	B–Commercial establishments
C–Automobiles	B–Bus terminals	C–Conference rooms
B–Banks	B–Cafeterias	C–Conventions
C–Bank counting rooms	C–Canneries	B–Corridors
C–Bank safe deposit departments	C–Central air conditioning systems	C–Creameries
	D–Chemical laboratories	C–Crowded rooms
		C–Dairies

TABLE A-2 (Continued)

C–Darkrooms
D–Decalcomania production
B–Department stores
C–Dentists' offices
B–Dining rooms
B–Display parlors
C–Distilleries
C–Doctors' offices
B–Drafting rooms
B–Dressing rooms
C–Drug stores
C–Dry cleaning plants
B–Educational institutions
B–Electrical installations
C–Elevators
D–Embalming rooms
C–Enclosed spaces
C–Engine rooms
B–Equipment rooms
C–Examination rooms
D–Exhaust hoods
C–Factories
B–Federal offices
C–Fermentation plants
D–Fertilizer plants
C–Fish markets
C–Five and Ten Cent Stores
C–Food processing
A–Forced air furnaces
C–Fruit storages
C–Funeral homes
C–Game rooms
D–Garbage disposal plants
A–Gravity return furnaces
B–Greenhouses
B–Grocery stores
C–Grills
A–Homes
C–Hospital rooms
C–Hospitals
B–Hotels
B–Hotel rooms
C–Incinerators
C–Individual cubicles
C–Industrial kitchens
B–Industrial offices
B–Institutions
B–Instrument rooms
B–Jewelry stores
C–Kitchens
D–Kitchen exhausts
D–Laboratories
C–Laundries
D–Leather processing
B–Libraries
D–Linoleum plants
C–Live poultry rooms
A–Living rooms
B–Lobbies
C–Locker rooms
B–Lounges
C–Lunch counters
C–Lunch rooms
B–Maintenance departments
C–Manufacturing plants
C–Mausoleums
D–Meat packing plants
C–Meat markets
C–Meat storage
B–Metal industries
C–Military equipment
B–Military installations
C–Mixed cold storage
C–Morgues
B–Motels
C–Motion picture studios
B–Municipal offices
B–Museums
D–New Processes
C–Night clubs
D–Nuclear processes
B–Nurseries
C–Odor barriers
B–Offices
C–Office buildings
C–Officers' clubs
D–Oilcloth production
C–Operating rooms
D–Paint departments
C–Paint plants
C–Penal institutions
C–Personnel protection
C–Pet shops
C–Pharmaceutical plants
C–Photo dark rooms
C–Photographic industry
C–Photographic studios
B–Planes
C–Plastics manufacturing
C–Plating shops
D–Pollution Control
C–Poultry processing
C–Poultry sales rooms
C–Prescription departments
C–Printing plants
B–Private offices
C–Processing laboratories
C–Processing rooms
D–Projection booths
C–Public assembly rooms
B–Public buildings
C–Public toilets
D–Pulp and paper plants
C–Radio studios
C–Railway cars
B–Railway stations
B–Reading rooms
B–Reception rooms
C–Recovery room, hospital
C–Recreation halls
C–Recreation rooms
C–Refineries
C–Refrigerated show cases
D–Rendering plants
C–Refrigerators
C–Research buildings
C–Reservation offices
A–Residences
D–Resin manufacturing
B–Restrooms
B–Restaurants
C–Restaurant kitchens
A–Retail shops
D–Rubber plants
B–Rumpus rooms
B–Sales rooms
B–Sample rooms
C–Schools
C–Service Departments
C–Sewage Disposal plants
D–Sewer vents
C–Show cases
C–Sick rooms
B–Silverware manufacture
C–Soap manufacture
B–Soda fountains
C–Specialty shops
B–State institutions
B–Steamships
B–Stock rooms
B–Storage spaces
B–Stores
C–Studios
B–Stuffy rooms
B–Super markets
C–Surgical rooms
B–Switchboard rooms
C–Tanneries
D–Tar processing
C–Taverns
C–Telephone booths
C–Telephone exchanges
C–Television studios
C–Test cubicles
B–Theaters
C–Theater lobbies
C–Theater lounges
B–Ticket booths
C–Toilets
B–Trains
B–Train reservation offices
C–Undertakers
B–Unit air coolers
C–Untidy rooms, hospital
C–Unventilated spaces
D–Varnish manufacture
C–Vegetable storage
D–Vent systems
C–Vestibules
C–Veterinary hospitals
B–Waiting rooms
C–Wards, hospital
B–Warehouses
D–Waste treatment plants
B–Window ventilators
B–Wood working plants
C–Work rooms
C–X-Ray darkrooms
B–Yachts
C–Youth clubs
C–Zoological gardens

DISCUSSION

WARREN VIESSMAN, Baltimore, Md. (WRITTEN): The author is to be complimented on presenting a very useful paper in regard to the application of activated charcoal adsorber equipment for the removal of a wide range of objectionable vapors and gases from the air. This means of air purification has been used and proven successful over a long period for quality control of cold storage fruits and for protection against war gases in

shelters and in gas masks. It would be of historical interest in this respect, if Mr. Barnebey could give us the date of the first modern mechanical application.

Use of activated charcoal for air conditioning has been retarded in the past by public psychological impressions and by unfamiliarity of the design engineer with the capability and method of effective application. The necessity of supplying fresh air for oxygen has been over emphasized. Tests have demonstrated that only about 0.89 cu ft of oxygen is required per man-hour. About 0.74 cu ft of carbon dioxide is liberated per man-hour for normal activity. More air is required to remove objectionable body odors than is required for removing carbon dioxide and for sustaining life in shelters or inclosures. Because of this, activated charcoal has become a very useful tool to the air-conditioning engineer. Fresh air need be supplied only to replenish the oxygen requirement and remove carbon dioxide. Odors and other objectionable gases and vapors can be removed by recirculation through activated charcoal within the inclosure.

Psychological objections are rapidly being overcome as regards odor removal. Activated charcoal to remove odors from toilet vents in submarines and recover the air is now in successful use.

Another application of recovery from toilet rooms is that of a 13-story office building in the Southwest. The building is air conditioned by heat pumps. Cooled air is supplied to the toilet rooms. As building utilization increased, a number of offices were created in the unconditioned basement. To provide air conditioning to this area and to minimize ventilation heat gain, the cooler air exhausted from the toilets was collected and sent through charcoal adsorbers to provide conditioned air to the basement offices. The air was free from objectionable odors and the spaces were cooler than before the system was installed.

For most air-conditioning applications, unimpregnated charcoals are adequate for atmospheric odor removal. Charcoals are impregnated to promote adsorption of a specific gas or vapor that is not readily adsorbed by unimpregnated charcoal. The impregnants either react directly with the gas being adsorbed or act as catalysts promoting physical adsorption, oxydation, decomposition, hydration, hydrolysis and perhaps reduction.

For the adsorption of ethylene given off in the cold storage preservation of plants, fresh foods and flowers, brominated charcoal is used. Activated charcoals for the removal of a number of war gases are usually impregnated with copper, silver and chromium salts.

Mr. Barnebey has effectively pointed out that there is a difference in activated charcoals. Retention and break-through time is affected. For the academic mind, the design of a charcoal adsorber is based on four concepts, known as *wave, capacity, two-layer, and diffusion and reaction.* These are explained and mathematical formulas presented for determining charcoal bed depth and pressure drop in an article on *How to Plan Air Conditioning for Protective Shelters,* appearing in the November and December 1954 issues of *Heating, Piping & Air-Conditioning.* The readily usable data presented by Mr. Barnebey will suffice for most designers of the usual installations.

In closing, I would like to place emphasis on the designer paying particular note to pressure drops as indicated by Mr. Barnebey for the various adsorbers and systems, as the fan will be required to overcome resistances in excess of those encountered for ordinary filters. For a particular adsorber the pressure drop will be materially affected by bed depth and air velocity.

S. F. DUNCAN, Los Angeles, Calif., (WRITTEN): Industrial applications of activated charcoal or carbon for the recovery or removal of vapors and the clarification and other treatments of liquors have been known for many years. Gas mask applications are also old and well known. The current trend toward cleaner air and the necessity for purification of ventilating air polluted by industrial waste, makes this paper most timely.

To the uninitiated, the sizing of activated charcoal filters has been more or less of a mystery. Certainly the chloropicrin test and rating is not widely comprehended. Mr. Barnebey has clarified these matters so that the practising engineer can understand the

reasons for a given selection of equipment even though he may not have all the data necessary to make the selection.

According to the literature, activated charcoal can be treated to have a preference for a particular gas. It has also been shown that passing 2 gases in sequence through activated charcoal may result in the release of the first absorbed gas and adsorption of the second. Generally the preference is for the larger molecules. In some cases this phenomenon has resulted in a slight odor being given off by an activated charcoal filter even though the overall performance was considered good. This was noted in tests run in Los Angeles and reported by Neal A. Richardson and Wilbur C. Middleton at the regional meeting of ASHAE, held in Los Angeles in May 1957. The paper was titled *Evaluation of Filters for Removing Irritants from Polluted Air.*

More information on the application and maintenance costs of activated charcoal filters will certainly be needed. This paper is a good first step along the path.

S. B. SMITH,† Pittsburgh, Penna., (WRITTEN): The paper is a worthwhile contribution which fills an obvious gap in the technical literature on this subject. His comprehensive and understandable treatment of a complex subject should be welcomed by the Society, especially since it offers several alternative approaches to problem solution.

However, as might be expected of a highly condensed presentation, several misleading inferences can easily be drawn from it; and these should be clarified.

Nature of Activated Charcoal: In general, the retentive character of charcoal is not critical in odor removal work; that is, most of the materials to be removed from the atmosphere are of the easily adsorbed type. In fact, the chloropicrin test suggested in the article is more a measure of total capacity for easily adsorbed material than for retentive character. Another commonly used test for adsorptive power that should be mentioned is the carbon tetrachloride capacity test. The result is usually reported as carbon tetrachloride picked up as a percent of the original charcoal weight after a dry air stream saturated with carbon tetrachloride at 0 C has passed through a column of carbon held at 25 C. The vapors are passed through the carbon column until no further weight gain is observed. Carbon tetrachloride capacities of 60–75 percent are not uncommon in commercial activated carbons. This determination is as suitable and is easier than the chloropicrin life test.

The alleged high tenacity of activated coconut charcoal is therefore not of particular value in most air-conditioning applications. Such carbons are, of necessity, harder to reactivate and are available only at premium cost. If the hardness and abrasion resistance are critically required, this cost may be justified.

Carbons from other sources, such as coal or wood, should definitely be considered since the same degree of retention may be obtained at more reasonable cost. A carbon of lower moisture affinity and greater resistance to ignition is sometimes required in critical applications. Contrary to a statement of the paper, coal-base carbons are made which have pores as fine and fully as retentive as those of activated coconut shell with only a slightly lower surface area per unit volume. In fact, not all grades of coconut (or other nut shell) carbons possess the fine pores described since their size is dependent on the method of activation.

Quantity of Activated Charcoal Required: In regard to alternative methods of estimating carbon requirements, perhaps more emphasis should be placed on the analytical approach (Method E). In addition to the checking of the effluent air from a test canister for odor, much can be done analytically on the exposed carbon:

1. The density of the carbon (or total weight increase) may be measured to within 2–5 percent to check the weight pick-up of impurities.
2. The carbon may be desorbed by heat and vacuum (or by steaming) and the desorbed vapors caught in a cold trap. After a separation of water, the quantity of material removed from the carbon may be measured and used as an estimate of the odor load to be removed by the adsorbent.

---

† Pittsburgh Coke & Chemical Company.

3. The material trapped in (2) may be analyzed by infrared spectrometry, gas chromatography, and/or mass spectroscopy.

At least one manufacturer of air pollution equipment has commonly employed such methods, and these will undoubtedly be used more extensively in the future, especially for potentially large installations. Such test services are available from commercial laboratories and, where used, serve not only to determine the adsorbable load but also to identify the principal contaminants.

Types of Activated Charcoal Purification Systems: It might be further emphasized that each optimum installation is a balance of equipment, maintenance and carbon replacement costs, adsorption efficiency and odor tolerance. In consideration of initial carbon costs or carbon replacement, the prime factor is the pounds of adsorptive capacity per dollar of carbon cost. Such factors as the volume occupied by the activated carbon and the resistance to air flow may vary from carbon to carbon, but these differences will be slight. From a practical point of view, a compromise from 70 percent CCl_4 capacity to 65 percent may afford a real saving without any loss in overall efficiency.

New Developments: It is soon evident in the economic calculations that the cost of the active component of the system, the activated carbon, is but a minor part of the total installation costs. Further, in replacement or reactivation of the adsorbent the handling and shipping costs are out of proportion to the value of the new or regenerated carbon itself. Because of the granular physical form of activated carbon, containers are complex, costly and difficult to service. The dense layer of carbon offers considerable resistance to air flow and layers must be placed in staggered or folded array to allow maximum cross-sectonal area to minimize head losses.

Answers to this problem are just beginning to appear in the form of disposable frames loaded (to a greater or lesser extent) with activated carbon. In some of these the amount of carbon is so small that they warrant little consideration; but some can be effective, provided the designer is assured that his calculated carbon requirement is met by such a unit. Obviously, if activated carbon could be obtained at attractive cost in self-supporting, loose mat form of low resistance, a greatly desired solution to odor removal would be at hand. Such products are now in the development stage in several locations and, when generally available, should be evaluated by the same criteria of efficiency and capacity as are the granular charcoals currently used.

HARRY BUCHBERG‡, Los Angeles, Calif.: I would appreciate it if the author would comment on a problem that arose when using activated charcoal to obtain clean air for air pollution studies.

Tests were run to determine the effectiveness of KI-2 activated charcoal in the removal of hydrocarbons from an air stream. Freeze-out air samples were taken before and after the filter bed and analyzed in a mass spectrometer. In one series of tests the inlet air was contaminated with a low concentration of propane gas. During subsequent tests it appeared that propane gas was being stripped off the charcoal. Could a small increase in the air temperature account for the stripping off of propane during periods of testing some time after tests with propane had been completed? Would the author please comment on the adsorptive power of activated charcoal in the removal of C_2, C_3, and C_4 hydrocarbons. Any data presented would be appreciated.

AUTHOR'S CLOSURE: Concerning the point raised by Mr. Duncan, when activated charcoal of suitable type is used for air purification it will not release ordinary odors when taking up higher molecular weight materials, provided the charcoal has not been used beyond the capacity recommended by the manufacturer.

Mr. Viessman asked about early uses, and I believe the first modern mechanical application of activated charcoal to air purification in living and working spaces was made about 1921 or 1922. It is difficult to pin down the exact date. Commercial manufacture of gas adsorbent charcoals was started in 1919 in the U. S. There is described in

‡ University of California.

the literature the use of ordinary charcoal for air purification in 1855. However, such charcoal had very low capacity for odors and its use was not practical.

Regarding the comments of Mr. Smith, the retentive character of activated charcoal is critical in odor removal work. Nothing is accomplished if odor is taken up and then not retained by the charcoal.

A wide variety of laboratory tests for estimating the capacity of activated charcoal for various vapors have been devised. A test should be used which closely approaches what is actually done in the use of the charcoal. The carbon tetrachloride test to which he refers is a capacity test to saturation. In other words, it indicates how much vapor a charcoal might take up no matter how much of this vapor passed the charcoal. The chloropicrin test mentioned is, however, a test to *break* which indicates how much vapor the charcoal might pick up before the first little bit passes through. This more nearly approaches what is desired in practice. It is possible to have types of activated charcoal which show up well by the carbon tetrachloride test but which are useless for odor removal. On the other hand, any charcoal which shows up good by the chloropicrin test to *break* is useful for odor removal.

Whether a small quantity of high grade adsorbent or a large quantity of low grade adsorbent is used for odor removal is, as Dr. Smith points out, a matter of economics. Since there is expense in changing the carbon and shipping it back and forth, high capacity has merit beyond the increase in the cost of the charcoal itself.

Mr. Buchberg asked about activated charcoal for removal of various hydrocarbons. C_2 hydrocarbons are adsorbed from air only slightly by activated charcoal at normal pressure and temperature. C_4 hydrocarbons are rather strongly adsorbed and charcoal has good capacity for them. C_3 hydrocarbons fall in-between and are border line cases. If activated charcoal has adsorbed propane to capacity at a particular temperature, a temperature increase would cause some of the propane to be released.

I appreciate the comments of the various discussers since they have raised excellent points, and which add to the scope of the paper.

1647

No. 1647

COOLING LOAD FROM PRETABULATED IMPEDANCES

By HARRY BUCHBERG*, LOS ANGELES, CALIF.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Department of Engineering, University of California, Los Angeles.

THE STATUS of the thermal network approach to the solution of heating and cooling problems arising in the air-conditioning industry may be summarized by listing the various techniques that can be employed to solve the network. Regardless of the technique of solution the starting point is the thermal network representation of the physical system and all of the limitations inherent in the network apply. The response of the network to periodic inputs has been computed by:

1. Solving the simultaneous node equations based on current continuity.[1]
2. Making direct measurements on an analogous electric network.[2]
3. Solving the differential equations written for the network using an electronic differential analyzer.[3, 4]

Solutions of the thermal network have also been obtained by summing the response of the network to a series of pulses representing transient inputs.[5]

Basically the 2 approaches taken to achieve a solution of the thermal network are on the one hand a summation of response to periodic steady state inputs and on the other hand the summation of response to pulse or transient inputs. The method of solution used in any instance depends on the following factors:

1. Objectives of the study being made and the number of solutions required.
2. Importance of transients in any particular design problem.
3. Particular computing experience and background acquired by available personnel.
4. Availability of particular types of computers.

For example, a design study to determine the optimum building characteristics including orientation, shading, surface treatments, location of openings, amount and location of thermal resistances and capacitances to minimize equipment operating costs will generally require a different computational procedure than the field determination of a cooling load. Furthermore, the type of computer available and particularly the previous experience of personnel undertaking the work in-

---

* Associate Professor of Engineering, Department of Engineering, University of California.
[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Minneapolis, June 1958.

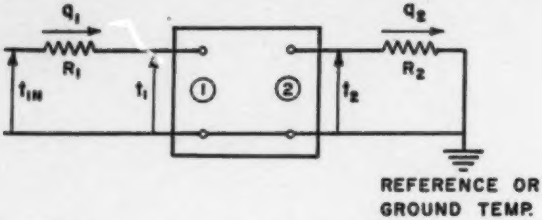


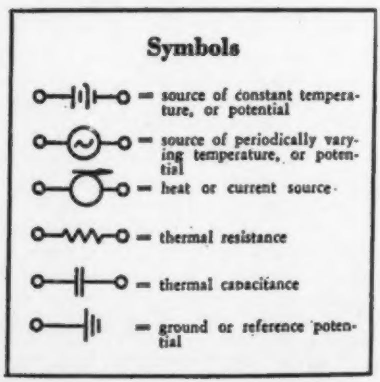
FIG. 1—CONDUCTION PATH NETWORK REPRESENTED AS A 4-TERMINAL NETWORK

fluences greatly the choice of computational method. One organization because of previous experience with a particular computer may find it very economical to proceed in a certain manner while another organization may find the same computational procedure to be very costly.

NEW METHOD PROPOSED

The opinion of the writer at this time is that direct measurements on an analogous electric network offers the greatest flexibility and economy for the treatment of design problems where many solutions of a network may be required to evaluate design parameters. On the other hand, for the determination of equipment operating loads, a computational procedure utilizing a maximum amount of previously computed and tabulated information is most desirable. With this objective in mind the complete thermal network representing an enclosed space was conceived of as a system of many 4-terminal networks in parallel, each 4-terminal network representing a heat conduction path into or out of the space as shown in Fig. 1.

Coupling of the networks at the inside and outside boundaries takes place through convection and radiation paths (see Fig. 2). If the inside space temperature is considered to be constant and uniform throughout the space it becomes convenient



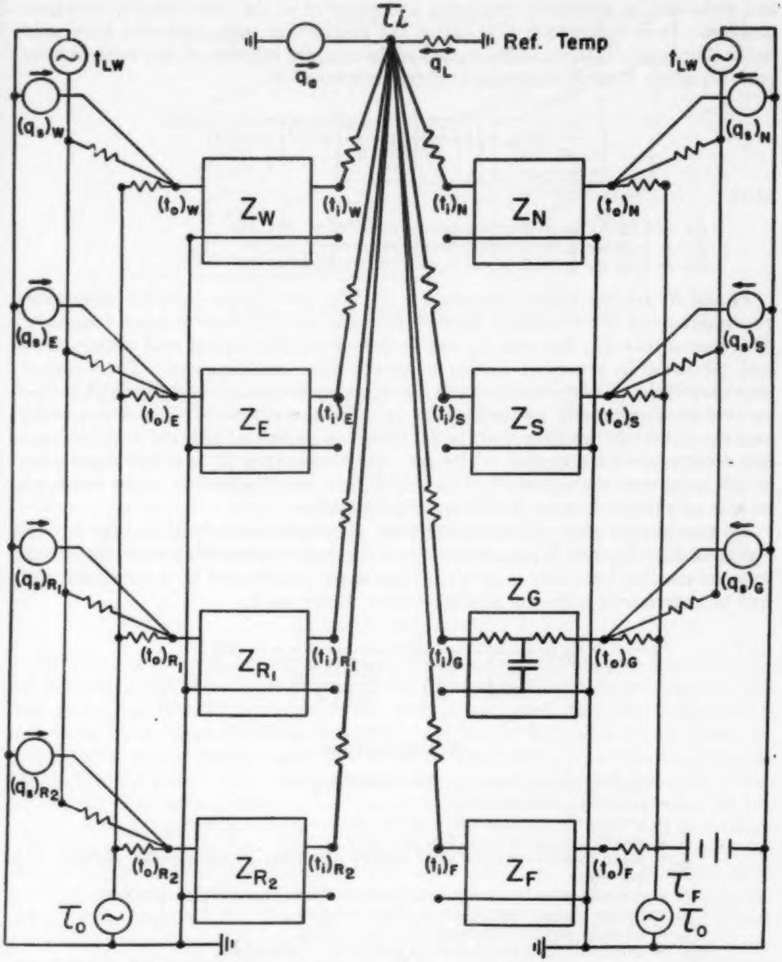


FIG. 2—THERMAL NETWORK REPRESENTING THE IDEALIZED TEST HOUSE WITHOUT INSIDE RADIATION EXCHANGE

to designate that temperature as the reference or ground potential and each of the 4-terminal networks are then coupled through the ground connection. Additional coupling occurs when radiation exchange is considered.

A solution of the complete thermal network consists of summing up the instantaneous contributions q_2 for all times of interest. The influence of heat sources

and sinks can be separately evaluated and included in the final solution by superposition. It is understood that all of the conduction path networks have only linear elements. If t_{in}/q_2 is designated as the transfer function of any heat transfer path Equation 1 can be demonstrated (see reference 6).

$$\frac{t_{in}}{q_2} = Z_{12} + \left[1 + \frac{Z_{11}}{R_1}\right]\left[1 - \frac{Z_{22}}{R_2}\right]\left[\frac{R_1 R_2}{Z_{12}}\right] \cdots\cdots (1)$$

where

$Z_{11} = t_1/q_1$ for $q_2 = 0$ (open circuit at terminal No. 2)
$Z_{22} = t_2/q_2$ for $q_1 = 0$ (open circuit at terminal No. 1)
$Z_{12} = t_1/q_2$ for $q_1 = 0$ (open circuit at terminal No. 1)

R_1 and R_2 are convective resistances. Z_{11}, Z_{22}, and Z_{12} are complex impedances characteristic of the 4-terminal network only and are functions of input frequency. This means that Z_{11}, Z_{22}, and Z_{12} can be determined for typical wall constructions and tabulated in a manner similar to steady state conductances. These impedances are entirely independent of the boundary resistances R_1 and R_2 which should be evaluated separately for each situation. For exterior walls t_{in} is time variable and dependent on weather conditions, radiation exchange with the surroundings, and solar radiation absorbed. The convective resistance R_1 is a function mainly of air speed over the exterior surface and R_2 is a function mainly of the difference in t_2 from the reference or inside space temperature.

To summarize: given a design input t_{in}, appropriate values of R_1 and R_2, and values of Z_{11}, Z_{22}, and Z_{12} characteristic of the construction being used, the instantaneous sensible heat flow in or out of the space contributed by a particular wall can be determined from the relation shown, Equation 2.

$$q_2 = \frac{t_{in}}{Z_{12} + \left[1 + \frac{Z_{11}}{R_1}\right]\left[1 - \frac{Z_{22}}{R_2}\right]\left[\frac{R_1 R_2}{Z_{12}}\right]} \cdots\cdots (2)$$

NOMENCLATURE

C = heat capacity, Btu per Fahrenheit degree.
q = heat flux, Btu per hour.
q_a = interior heat source, Btu per hour.
q_l = sensible cooling load, Btu per hour.
q_s = solar input or direct solar energy absorbed by an exposed surface, Btu per hour.
R = resistance to heat transfer, Fahrenheit degrees per Btu per hour.
t_i = inside surface temperature, Fahrenheit.
t_o = outside surface temperature, Fahrenheit.
t_{LW} = outside long-wave radiation potential, Fahrenheit.
Z = conduction path thermal impedance, Fahrenheit degrees per Btu per hour.

Greek Letters

θ = time.
τ_i = inside ambient air temperature, Fahrenheit.
τ_o = outside ambient air temperature, Fahrenheit.
τ_F = under floor ambient air temperature, Fahrenheit.

Subscripts

E = east wall.
F = floor.

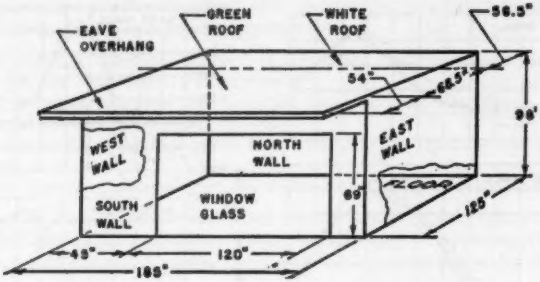


Fig. 3—Schematic Diagram of Test House

This kind of computation can be made by using a hand calculator once tabulations are made of the characteristic complex impedances. Only 3 tabular entries, Z_{11}, Z_{22}, Z_{12}, have to be made for each type of wall section providing sufficient accuracy is obtained through representing t_{in} as the sum of a d-c component and the fundamental sinusoid only. Additional harmonics would require additional entries, 3 for each frequency. The complex impedances Z_{11}, Z_{22}, and Z_{12}, can be determined by direct measurement on model sections, by measurements made on an analogous electric network (network computer), or by various analytical approaches.

Example Solution

Because the complex impedances are functions of input frequency, it is desirable to determine the importance of harmonics in an actual load determination. In this study, the transfer functions of the various wall, roof, and floor sections of a one-room wood frame dwelling (described in detail in Reference 2 and shown in Fig. 3) were determined by Laplacian transformation methods applied to a lumped *RC* ladder network. The thermal network representing the test house is shown in Fig. 2. Two sets of transfer functions (one for daytime conditions and one for night) were computed for each structural section except the east wall and floor. This was necessary to take into account the change in boundary resistance from day to night because of different average wind velocities. The conduction path networks, values of resistance and capacitance, and transfer functions computed are given in Appendix A. Contributions to the sensible cooling load by each of the structural sections and the resulting total cooling load as a function of time were computed using the sol-air inputs based on test house measurements made during a 24-hr period beginning September 8, 1953.[2] The sol-air inputs were computed actual inputs used on the d-c network computer when solutions were obtained for comparison. The sol-air inputs computed for the various orientations are given in Appendix B.

Results and Discussion

The results of the hand computations are presented in Figs. 4 through 6. Three determinations of heat fluxes and sensible cooling load as a function of time of

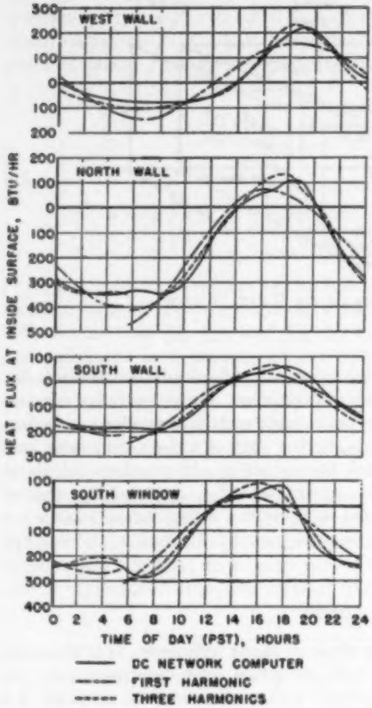


FIG. 4—HEAT FLUX CURVES FOR SOUTH, NORTH, AND WEST WALLS

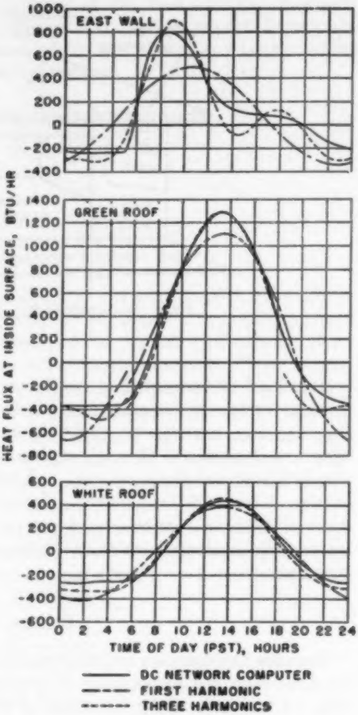


FIG. 5—HEAT FLUX CURVES FOR ROOF SECTIONS AND EAST WALL

day are presented, *viz*: the d-c network computer prediction, the prediction based on the fundamental input only, and the prediction based on the first three harmonics. The discontinuities in the heat flux and load curves resulted from the shift in boundary resistance from night to day and day to night values. The cooling load is based on maintaining a constant space temperature of 76 F. The heat source, q_A, was constant with time and equal to 1280 Btu per hr.

Examination of the separate wall heat flux curves in Figs. 4 and 5 indicates that the peak load prediction based on the fundamental input only is lower than the more accurate d-c network solution in all cases. The greatest deviation occurred for the East wall. The attenuation of the higher harmonics due to the insulation is apparent from the West wall curves. Due to lack of insulation in the East wall one would expect the higher harmonics to exert more influence for this orientation in this example. A more complete study of the importance of input harmonics in steady state sinusoidal solutions for various typical constructions has been completed and will be written up for publication in the near future.

Fig. 6 presents the predictions of sensible cooling load as a function of time of day. Four predictions are shown for comparison:

1. The d-c network computer solution in which the input functions were accurately represented and the load directly determined.
2. The d-c network computer solutions for the individual heat fluxes and the heat source, summed by hand to obtain the load.
3. The steady-state sinusoidal prediction based on the fundamental input or first harmonic only.
4. The steady-state sinusoidal prediction based on the first 3 harmonics.

It can be seen that the prediction based on the fundamental input only compares very well with the direct analogue solution, in this case even better (in the region of the peak load) than the prediction based on representation by 3 harmonics. Harmonics appear to be much more important during the night hours which becomes apparent when examining the input curves. The 3 determining factors in assessing the importance of harmonics in any situation appear to be:

1. Shape of the input functions,
2. Degree of attenuation of the higher harmonics by the RC network representing the conduction path, and
3. Size of the individual heat flux contributions compared to the total instantaneous load.

Comparison between predictions (1) and (2) (given in Fig. 6) give some idea of the consistency of solutions with the d-c network computer. The differences in the curves arise from errors in reading the tapes for the separate heat fluxes.

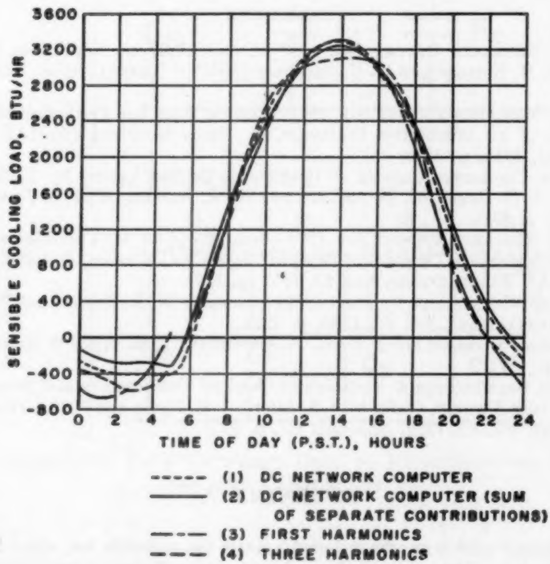


FIG. 6—COMPARISON OF COOLING LOAD CURVES

It should be pointed out that one of the greatest limitations of the steady-state sinusoidal method is the difficulty of handling an inside radiation exchange network. The writer has thus far not been able to find a simple computational procedure other than usual reiterative methods. The network solved in this study did not include inside radiation exchange. Another procedure that can be followed is to somehow adjust the inside boundary resistances to include radiation in the manner used in steady state calculations. In the writer's opinion, the best treatment of inside radiation exchange remains the incorporation of an inside resistance coupling network and solution with a d-c network computer.[2] Steady-state radiation sources, however, can be handled simply with the steady-state sinusoidal method.

Subsequent to these studies it has been found that the most economical procedure for applying the sinusoidal steady-state method is to determine the transfer functions, t_{in}/q_2, directly on an a-c network computer and tabulate these data for typical constructions and various values of boundary resistance rather than tabulating the characteristic transfer and mutual impedances Z_{11}, Z_{22}, Z_{12}. This will reduce considerably the labor in making design calculations. A paper describing these subsequent studies is now in process.

ACKNOWLEDGMENTS

The assistance of undergraduate students in many of the calculations is gratefully acknowledged. The author is also indebted to Dr. Don Lebell for several suggestions involving network theory.

REFERENCES

1. ASHVE RESEARCH REPORT No. 1497—Circuit Analysis Applied to Load Estimating, by H. B. Nottage and G. V. Parmelee (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 59).
2. ASHAE RESEARCH REPORT No. 1543—Electric Analogue Prediction of the Thermal Behavior of an Inhabitable Enclosure, by Harry Buchberg (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 339).
3. Analogue Computer Analysis of Residential Cooling Loads, by T. N. Willcox, C. T. Oergel, S. G. Reque, C. M. toeLaer and W. R. Brisken (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 505).
4. ASHAE RESEARCH REPORT No. 1595—Analysis of an Air Conditioning Thermal Circuit by an Electronic Differential Analyzer, by G. V. Parmelee, P. Vance and A. N. Cerny (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 129).
5. Heat Load Calculations by Thermal Response, by W. R. Brisken and S. G. Reque (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 391).
6. *Introductory Circuit Theory*, by E. A. Guillemin (John Wiley & Sons, Inc.,New York, 1953, p. 157).
7. Effect of Heat Storage & Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan, and F. O. Urban (ASHVE TRANSACTIONS, Vol. 45, 1939, Appendix III, p. 393).

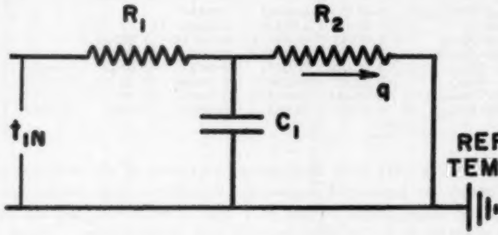
APPENDIX A

The conduction path networks included a single tee, a double tee, and a triple tee as shown in Fig. A-1. The values of thermal resistance and capacitance used in the solutions are given in Table A-1. The transfer impedances t_{in}/q for each of the networks

BUILDING COMPONENT	CONDUCTION PATH NETWORKS

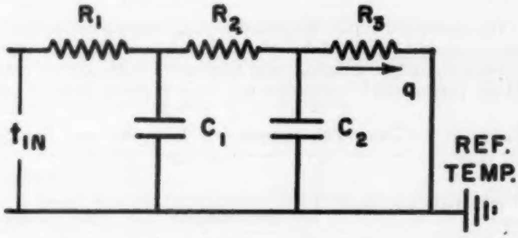
1. SINGLE TEE

SOUTH WINDOW FLOOR



2. DOUBLE TEE

ROOF SECTIONS (White & Green)



3. TRIPLE TEE

WALLS
West
East
North
South

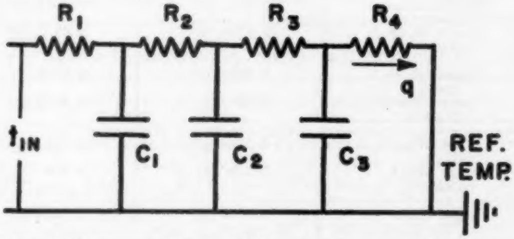


Fig. A-1—Conduction Path Networks Used to Represent the Different Building Components

TABLE A-1—VALUES OF THERMAL RESISTANCE AND CAPACITANCE FOR ALL BUILDING COMPONENTS

STRUCTURAL COMPONENTS	RESISTANCE, FAHRENHEIT PER BTU (HR)					CAPACITANCES, BTU PER F DEG		
	R_1 (DAY)	R_1 (NIGHT)	R_2	R_3	R_4	C_1	C_2	C_3
SOUTH WINDOW	0.01187	0.0285	0.0459			17.7		
FLOOR	0.0238	0.0238	0.0134			171		
WHITE ROOF	0.00800	0.00875	0.00768	0.00560		12.5	77.7	
GREEN ROOF	0.00725	0.01275	0.00635	0.00485		14.0	94.0	
WEST WALL	0.01150	0.0251	0.0737	0.0700	0.0309	90.8	32.4	29.6
EAST WALL	0.0152	0.0152	0.01056	0.00675	0.0309	90.0	0.288	28.3
NORTH WALL	0.00534	0.01372	0.00458	0.00713	0.0254	94.0	0.416	133
SOUTH WALL	0.00975	0.0248	0.00845	0.0131	0.0449	48.2	0.344	72.6

shown in Fig. A-1 were determined in terms of the resistances, capacitances, and input frequency by standard Laplacian transformation methods. For each structural section, the appropriate values of C and R were substituted and the transfer impedance was reduced to an amplitude ratio and phase angle. These are given in Table A-2 for each structural section and for the first 3 input harmonics.

APPENDIX B

The combined input temperature (t_{in}) commonly known as the sol-air temperature was computed for all of the orientations by summing up the input contributions due to convection, solar radiation, and long-wave radiation exchange. The hourly values of sol-air temperature computed for each exposed surface are given in Table B-1. A

TABLE A-2—TRANSFER IMPEDANCE FACTORS FOR ALL STRUCTURAL COMPONENTS

BUILDING COMPONENT	TRANSFER IMPEDANCE, t_{in}/q		
	1ST HARMONIC	2ND HARMONIC	3RD HARMONIC
SOUTH WINDOW (DAY)	0.0576 /2.5°	0.0576 /5°	0.0581 /7°
SOUTH WINDOW (NIGHT)	0.0741 /5°	0.0750 /9°	0.0763 /14°
WHITE ROOF (DAY)	0.0744 /15°	0.0813 /29°	0.0913 /40°
WHITE ROOF (NIGHT)	0.0750 /16°	0.0825 /30°	0.0938 /41°
GREEN ROOF (DAY)	0.0645 /16°	0.0706 /30°	0.0805 /41°
GREEN ROOF (NIGHT)	0.0675 /4°	0.0679 /8°	0.0663 /13°
WEST WALL (DAY)	0.215 /47°	0.296 /84.7°	0.435 /113°
WEST WALL (NIGHT)	0.218 /53.2°	0.390 /84.1°	0.515 /108°
EAST WALL	0.0586 /16°	0.0616 /31°	0.0688 /45°
NORTH WALL (DAY)	0.0466 /26°	0.0545. 47°	0.0672 /63°
NORTH WALL (NIGHT)	0.0590 /37°	0.0792 /62°	0.1074 /80°
SOUTH WALL (DAY)	0.0831 /23°	0.0975 /39°	0.1138 /48°
SOUTH WALL (NIGHT)	0.1060 /31°	0.1338 /47°	0.158 /55°

TABLE B-1—HOURLY SOL-AIR TEMPERATURES FOR ALL EXPOSED SURFACES

TIME HOURS	SOL-AIR TEMPERATURE[a], FAHRENHEIT—76 DEG						
	EAST WALL	WEST WALL	NORTH WALL	SOUTH WALL	SOUTH WINDOW	GREEN ROOF	WHITE ROOF
0	−16.81	−20.89	−19.37	−18.41	−17.33	−24.41	−28.05
0100	−16.75	−20.53	−19.28	−18.55	−17.27	−24.41	−28.05
0200	−15.69	−19.28	−18.18	−17.30	−16.30	−23.69	−26.78
0300	−15.19	−18.69	−17.64	−16.70	−23.11	−25.48	−15.74
0400	−14.69	−18.72	−17.64	−16.70	−23.11	−25.48	−15.80
0500	−16.10	−19.56	−18.62	−17.73	−23.96	−27.27	−16.77
0600	+59.15	−15.29	−14.11	−15.08	−12.78	− 3.85	−15.04
0700	+61.0	−15.10	−15.07	−14.85	− 2.82	+19.03	−15.90
0800	+52.14	−13.76	−14.06	−13.74	+ 6.74	+38.78	−15.32
0900	+40.50	−11.19	−11.70	−11.25	+15.11	+55.10	−13.29
1000	+27.16	− 5.84	− 6.96	− 6.42	+24.15	+69.65	− 8.80
1100	+13.12	− 0.96	− 2.34	− 1.94	+30.05	+78.20	− 4.37
1200	+ 1.47	− 6.48	− 1.60	− 2.01	+34.60	+83.70	− 0.41
1300	+ 3.86	−24.05	+ 3.89	+ 4.25	+34.73	+80.87	+ 1.98
1400	+ 3.66	+37.10	+ 4.06	+ 4.34	+30.64	+72.60	+ 2.40
1500	+ 2.85	+47.70	+ 4.13	+ 4.24	+24.27	+57.73	+ 2.83
1600	+ 2.59	+54.64	+ 4.61	+ 4.60	+16.67	+39.81	+ 3.74
1700	+ 2.30	+59.53	+ 5.60	+ 5.40	+ 6.40	+15.71	+ 5.11
1800	+ 1.87	+57.98	+12.12	+ 5.31	− 3.90	− 6.83	+ 5.01
1900	− 2.51	−12.05	− 5.65	− 4.47	−11.86	−19.80	− 2.67
2000	−10.00	−17.44	−13.05	−11.89	−18.09	−25.36	−10.18
2100	−14.35	−20.17	−17.09	−15.93	−21.64	−27.90	−14.44
2200	−15.57	−20.70	−18.34	−17.22	−23.09	−28.53	−15.90
2300	−16.44	−21.88	−19.10	−18.05	−24.03	−28.42	−16.85

[a] Sol-air temperature inputs are reckoned wth respect to a reference temperature of 76 F.

Fourier analysis was made of each of the sol-air temperature-time functions in accordance with the procedure outlined in Reference 7. Table B-2 presents the Fourier Series functions with 3 harmonics representing each of the surfaces.

DISCUSSION

G. V. PARMELEE, Dhahran, Saudi Arabia, (WRITTEN): The need for simple methods of handling the thermal capacity of the interior structure and contents of an air-conditioned building has been brought out in every paper on the subject of thermal circuits. Each paper has given a little more understanding of how elements in a circuit

TABLE B-2—SOL-AIR TEMPERATURE-TIME FOURIER SERIES FUNCTIONS

Structural Component	Sol-Air Temperature-Time Fourier Series Functions
East Wall	$4.90 + 24.6\cos(\omega_0\theta - 142^\circ) + 17.2\cos(\omega_1\theta + 135^\circ) + 11.6\cos(\omega_2\theta - 14^\circ) + \cdots\cdots$
West Wall	$0.64 + 32.1\cos(\omega_0\theta + 133^\circ) + 18.6\cos(\omega_1\theta - 112^\circ) + 9.61\cos(\omega_2\theta - 17^\circ) + \cdots\cdots$
North Wall	$-8.84 + 13.04\cos(\omega_0\theta + 141^\circ) + 3.97\cos(\omega_1\theta - 112^\circ) + 1.94\cos(\omega_2\theta - 85^\circ) + \cdots$
South Wall	$-8.59 + 12.38\cos(\omega_0\theta + 142^\circ) + 3.57\cos(\omega_1\theta - 101^\circ) + 1.42\cos(\omega_2\theta - 94^\circ) + \cdots$
South Window	$-8.80 + 11.1\cos(\omega_0\theta + 134^\circ) + 3.72\cos(\omega_1\theta - 108^\circ) + 1.42\cos(\omega_2\theta - 107^\circ) + \cdots$
White Roof	$-1.56 + 30.2\cos(\omega_0\theta + 171^\circ) + 6.98\cos(\omega_1\theta - 12^\circ) + 1.17\cos(\omega_2\theta - 24^\circ) + \cdots$
Green Roof	$12.89 + 58.0\cos(\omega_0\theta + 176^\circ) + 16.3\cos(\omega_1\theta - 2^\circ) + 3.56\cos(\omega_2\theta + 4^\circ) + \cdots$

interact and the solutions of problems have produced the magnitudes of the storage effect for specific situations. It is evident that a great deal of work must be done before engineers have a comprehensive body of data which can be applied to their everyday problems. This discusser has often thought that it might be possible to develop constants which could be applied to instantaneous rates of heat flow to yield cooling load. Sometime ago he needed some such factors and looked into the matter instead of just thinking about it. It seems worthwhile and appropriate to the discussion of Mr. Buchberg's paper to give some of the results of this investigation.

The basic concept in the development is the fact that the diurnal cooling load is the sum of a 24-hr average rate of heat flow and a periodic component. It is the periodic component that is damped by the thermal capacity of a building and it seems reasonable that a constant factor could be applied to this component. Cooling load would then be the sum of the 24-hr average rate of heat flow plus the reduced periodic component. In 1948, J. P. Stewart† showed that the instantaneous rate of heat flow through a wall or roof could be estimated with reasonable accuracy by Equation A.

$$q = UA(t_m - t_i) + \lambda_1 UA(t_e - t_m), \text{ Btu per hr} \quad . \; . \; . \; . \; . \quad \text{(A)}$$

where

U = the overall coefficient of heat transfer, Btu per (hr) (sq ft) (F deg)
A = area, sq ft
t_m = 24-hr average sol-air temperature, F deg
t_i = the constant indoor temperature, F deg
t_e = the sol-air temperature, F deg, at an hour earlier than the hour in question by the time lag of the wall or roof
λ_1 = a decrement factor, dimensionless

The decrement factor, λ_1, is made up of parts of the fundamental and the second harmonic decrement factors as found in the charts prepared by Mackey and Wright‡. The quantity $(t_e - t_m)$ is roughly the same as the fundamental harmonic of the temperature wave. The time lag is the sum, approximately, of the fundamental time lags of each material in the wall or roof. Values appropriate to different orientations and structure weights were last published in the 1952 GUIDE.

The procedure was to develop an additional constant decrement factor for the second term of this equation to account for the damping of the internal structure. The additional time lag was taken into account in the selection of t_e in the second term. If one examines the results of circuit problems that have been published recently, it will be seen that the 24-hr average cooling load is always less than the 24-hr average input. This is because in an air-conditioned room, the interior room surfaces are always warmer than the room air, if the outdoor air is above the indoor air temperature. The effect of this is to diminish the rate of heat inflow through the exterior parts of the building (or to aid in the escape of sensible heat generated by internal sources, such as lights) compared to a condition of equal indoor air and surface temperatures. Hence, a third decrement factor needs to be applied to each part of Equation A. This equation, modified, for walls and roofs becomes;

$$q = \lambda_3 UA(t_m - t_i) + \lambda_1\lambda_2\lambda_3 UA(t_e' - t_m), \text{ Btu per (hr)} \quad . \; . \; . \quad \text{(B)}$$

For glass transmitting solar radiation, the equation is:

$$q = \lambda_3 A(I_m\tau_m) + \lambda_2\lambda_3 A(I_e'\tau_e' - I_m\tau_m) \quad . \; . \; . \; . \; . \; . \; . \quad \text{(C)}$$

where

t_e' = sol-air temperature, deg F, at an hour earlier than the hour in question by the time lag of the wall or roof plus the additional time lag due to the heat capacity of the interior building structure

† Solar Heat Gain Through Walls and Roofs for Cooling Load Calculations, by J. P. Stewart (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 361)
‡ ASHVE RESEARCH REPORT No. 1255—Periodic Heat Flow—Homogeneous Walls and Roof, by C. O. Mackey and L. T. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 50, 1944, p. 293).

TABLE A—COMPARISONS OF COOLING LOADS DETERMINED BY CONSTANT LOAD DECREMENT FACTORS WITH COOLING LOADS DETERMINED BY CIRCUIT ANALYSIS

Time	Example 1[a]				Example 2[b]		
	Input Function	Load by Thermal Circuit	Load by Constant Load Decrement Factor		Input Function	Load by Thermal Circuit	Load by Cons. Load Dec. Factor Method A
			Method A	Method B			
12 mid		204	193	156	77 F	2.3	1.8
2 am		126	193	105	76	1.2	1.2
4		84	193	156	74	0.5	1.0
6	905	205	193	294	75	0.3	0.6
8	891	500	264	483	80	0.5	0.8
10	1040	643	628	672	87	1.2	1.7
12 noon	1140	783	716	810	106	2.5	2.9
2 pm	1040	824	793	861	126	5.8	6.2
4	891	823	793	810	127	9.3	9.8
6	905	757	716	672	106	9.7	9.9
8		500	628	483	85	6.8	6.2
10		264	264	294	81	4.3	2.6
Ave. 24-hr.	537	483	478	485	91.6	3.7	3.7
λ_1			0.60	0.70			1.03
λ_2			0.90	0.90			0.90

Example 1—Solar radiation transmitted through north window glass, Btu per hr.
[a] ASHAE RESEARCH REPORT 1529—Circuit Analysis Applied to Load Estimating, Part II—Influence of Transmitted Solar Radiation, by H. B. Nottage and G. V. Parmelee (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 125).
Example 2—Sol-air temperature on southwest wall of an air and panel-cooled room, Btuh sq ft floor.
[b] ASHAE RESEARCH REPORT 1595—Analysis of an Air-Conditioning Thermal Circuit by an Electronic Differential Analyzer, by G. V. Parmelee, P. Vance, and A. N. Cerny, (ASHAE TRANSACTIONS, Vol. 67, 1957, p. 129).

$I_m\tau_m$ = 24-hr average transmitted solar radiation, Btu per (hr) (sq ft)
$I_e'\tau_e'$ = solar radiation transmitted at an hour earlier than the hour in question by the time lag due to the heat capacity of the interior building structure, Btu per (hr) (sq ft)
λ_2 = decrement factor due to the damping effect of the interior building structure, dimensionless
λ_1 = decrement factor due to inequality of room air and surface temperatures dimensionless

The two decrement factors were found by comparing hourly values of cooling load, as determined by thermal circuit analysis for a given situation, with corresponding hourly values of instantaneous rate of heat flow. It is hardly necessary to state that there were few circuit results available for this comparison. Furthermore, only those results which gave loads as the result of a single input function could be used. The diurnal cooling load was re-expressed as a 24-hr average plus a diurnal periodic component. This periodic component was found simply by subtracting the 24-hr average load from each hourly value of cooling load. The periodic component of the diurnal instantaneous rate of heat flow was found in 2 different ways. In Method A, the 24-hr average rate was subtracted from the hourly values of instantaneous rate of heat flow. (These are given in tabular form in THE GUIDE.) This component can of course be computed directly from the second member of Equation A. In Method B, the fundamental harmonic was taken as the periodic component. Mr. Buchberg indicated in his present paper that the fundamental yields the instantaneous rate of heat flow with satisfactory accuracy.

The ratio of the 24-hr average cooling load to the 24-hr average instantaneous rate of heat flow gave the decrement factor, λ_2. Next, hourly values of λ_1 were computed

by dividing the load periodic component by λ_2 times the corresponding instantaneous heat gain periodic component. The latter was shifted time-wise so that the peak cooling load and peak instantaneous heat gain coincided. Hourly ratios were averaged to give λ_2. As might be expected, the ratios varied considerably from hour to hour. The variations were greatest at the 2 points where the sign of the periodic component reversed. At these points, the ratios were usually anomalous values.

Table A compares the results of using both methods with the load determined by circuit analysis for solar radiation transmitted through window glass. In spite of its crudity, Method A gives a fairly satisfactory representation of cooling load over most of the period of principal interest. Method B is better, especially during the period of low rates. Table A also gives the Method A results for the load due to a sunlit southwest wall in a room cooled by air plus a cool ceiling panel. The result of λ_2 being greater than unity has no significance. Better agreement than in the first case was expected because the function in this example is continuous rather than discontinuous. This may be an effect of the cool panel.

These examples are insufficient to form any judgement, but do suggest that the development of constant load decrement factors, as they might be called, is worth some further investigation. In this connection it is strongly urged that in future work in circuit analysis, cooling loads should be reported not only as totals but also as individual components. Perhaps this discussion will suggest a useful way of making circuit results applicable to practical load-estimating problems.

In this latest paper of a series, Mr. Buchberg has omitted radiation exchange between interior room surfaces. Hence, the cooling loads do not completely reflect the thermal capacity of the interior structure. In his first paper* the cooling load shown by Fig. 11 is quite similar to that shown by the solid curve of Fig. 6 of the present paper. Apparently this is because the structure is a lightweight one. I would appreciate Mr. Buchberg's comments on these 2 sets of results.

AUTHOR'S CLOSURE: Mr. Parmelee has taken this opportunity to present an empirical method of load calculation which attempts to account for the thermal capacity of interior structure and contents of a building. My only comment at this time is that the method presented appears to lack a good theoretical basis. It is rather an empirical method which attempts to determine constants for an equation by fitting it to limited data obtained from previous thermal network solutions. Its validity is, therefore, limited to identical examples.

It has been our experience that the most satisfactory method of analyzing the building system as a whole is by solving the representative thermal network on a d-c network computer. The transient response method presented by Brisken also has considerable merit in certain cases.

In answer to Mr. Parmelee's question regarding the similarity of the solid curve in Fig. 6 with Fig. 11 in a previous publication, it would be much more valid to compare curves C-6 and C-2 in Fig. 5 of *Cooling Load from Thermal Network Solutions* published in the ASHAE JOURNAL SECTION, October 1957 to determine the importance of including the inside radiation exchange network for this example. Neglecting the radiation exchange between interior surfaces reduced the predicted peak instantaneous load by about 15 percent and the total daily load by about 9 percent. Comparisons with Fig. 11 as suggested by Mr. Parmelee are not valid because a different network configuration was used for that solution and some of the circuit parameters were subsequently changed.

More experience with the importance of including the inside radiation exchange network is needed. The TAC on Thermal Circuits has just suggested that additional work on this problem be done.

---

* ASHAE RESEARCH REPORT No. 1543—Electric Analogue Prediction of the Thermal Behavior of an Inhabitable Enclosure, by Harry Buchberg (ASHAE TRANSACTIONS Vol. 61, 1955, p. 339).

1648

No. 1648

SOLAR ENERGY UTILIZATION FOR HEATING, COOLING, DISTILLATION AND DRYING

This paper is a technical contribution made by the members* of the Technical Advisory Committee on Solar Energy Utilization, who are its joint authors.

THE AVAILABILITY of energy and its adaptation to the accomplishing of useful ends is the most important single factor in determining the standards of living of peoples and nations. Currently, however, the vast majority of the energy resources utilized in producing power and heat are derived from fossil fuels. Unfortunately, these stored energy resources are limited in reserve and unequally distributed. The annual world energy consumption for all purposes is equivalent to approximately 38 billion tons of bituminous coal annually. Yet it is projected that the increasing population of the world coupled with the desire for higher standards of living may increase this rate of energy consumption as much as 50 times within the next century.

The world must, therefore, look to other comparatively untapped sources of energy and power if civilization is to continue its technical development and if the underdeveloped countries are to be brought to their fullest potential. Fortunately, two vast sources of energy remain virtually untapped, and their availability depends to a great extent upon steady technical progress as well as technological breakthroughs. One is the harnessing of fission and fusion processes and the other is the gathering and utilization of solar energy.

The uses to which solar energy are most likely to be put will vary between countries depending upon the energy availability and economic status. In countries such as Sweden and the United States the most likely initial uses appear to be in the field of space heating. In the warmer countries such as India and Israel the adaptation of solar energy to water heating, drying and refrigeration will be of great and probably initial importance. Solar energy developments in the fields

---

* They are: R. C. Jordan, USA, Chairman; F. R. Ellenberger, USA; M. L. Ghai, USA; H. C. Hottel, USA; N. B. Hutcheon, Canada; M. L. Khanna, India; G. O. G. Löf, USA; C. O. Mackey, USA; Gunnar Pleijel, Sweden; Maria Telkes, USA; J. L. Threlkeld, USA; G. T. Ward, Singapore; and L. F. Yissar, Israel.

Presented at the Semi-Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Minneapolis, June 1958.

of distillation and pumping are adaptable to many areas of the world where water must be purified and moved.

In 1955, in recognition of the potential importance of solar energy utilization to the members of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, the Society formed a Technical Advisory Committee on Solar Energy Utilization. Because of the world-wide interest in this activity the committee was made international in scope with membership from Malay, Israel, Sweden and India as well as the United States and Canada. The scope of the committee was defined as the development of basic and applied information pertinent to the use of solar energy for heating, cooling, distillation and drying. In order to best outline and survey the research necessary in this field, the committee elected to prepare for the Society the outline and discussion here presented. It is not in-

SOLAR COLLECTOR EXPERIMENTS IN SINGAPORE, MALAYA

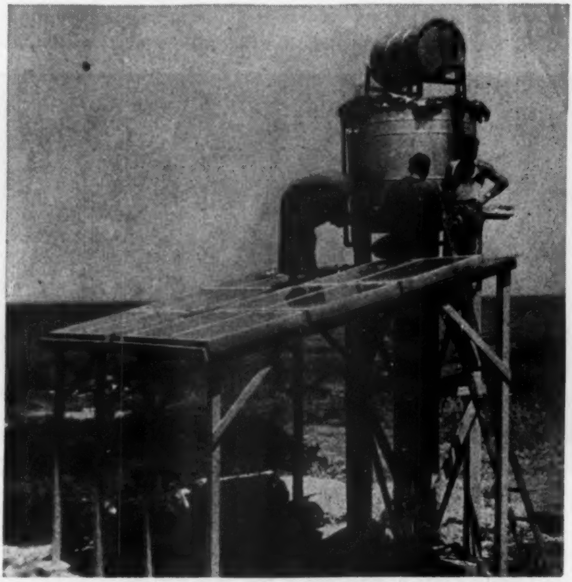
tended that this outline suggest directly specific problems which need research activity but rather to outline broadly the technical information needed in the solution of solar utilization problems. In some cases the information suggested is already available, while in others technical information may be completely lacking. It is advised that those considering the undertaking of researches in the solar utilization field thoroughly survey the source literature available.

The most complete bibliography of significant literature in this field is to be found in the book *Applied Solar Energy Research* published in 1955 by the *Stanford Research Institute* for the *Association for Applied Solar Energy*. Since that time the best single reference of recent solar utilization literature is to be found in *The Journal of Solar Energy, Science and Engineering* as published by the *Association for Applied Solar Energy*, Central Plaza Bldg, 3424 N. Central Ave., Phoenix, Ariz. However, many other technical journals such as the JOURNAL SECTION of the Society, published in *Heating, Piping & Air Conditioning*, have provided original research contributions and such literature should also be surveyed.

In the accompanying outline, Section 1—The Availability of Solar Radiation, Section 2—Solar Collectors and Section 3—Solar Heat Storage, are basic to all utilization problems. The solar source is a transient one, varying not only seasonally and geographically but also with such meteorological variables as cloud

pattern, atmospheric contamination and atmospheric water vapor content. There is at present insufficient knowledge to permit even a statistical prediction of the availability of solar energy in most areas of the world. This is the starting point for all solar utilization problems.

Collection of solar energy is the next problem. Both moving and fixed collectors have obvious disadvantages. For high-temperature collection, concentrating devices are necessary, but for most low-temperature applications, plane collectors



A CUSTOM BUILT SOLAR HOT WATER HEATER IN USE IN ISRAEL AS AN EXAMPLE OF THE APPLICATIONS OF SOLAR ENERGY WHICH ARE BEING MADE IN MANY PARTS OF THE WORLD

would appear to have dominant advantages. Potentially, improvements in collection either in cost or efficiency may be possible through such technical considerations as wavefront and wavelength discrimination, the use of translucent solar trapping materials other than glass, or through simplified combinations of collector surfaces and configurations or transport passages. In the final analysis, if collector systems can be constructed and maintained cheaply enough, a vast number of solar utilization devices become immediately practical.

The third area of basic consideration concerns solar heat storage. This again in final analysis is an economic problem. In some solar applications such as the pumping and distillation of water the end effect can be stored and thermal storage

in itself is not necessary. However, in applications of solar energy to space heating and cooling, thermal storage becomes necessary, since spaces must be maintained at predetermined temperature levels rather than permitting them to be overheated or overcooled and provision made for continuous service during sunless hours.

Section 4—Architectural Considerations, involve those factors which must be taken into consideration when solar collectors and storage systems are to be integrated into architectural structures. For example in the heating and cooling of buildings micrometeorological variations and the obstructing effects of the surrounding terrain and of surrounding buildings must be thoroughly considered. In

AN EXAMPLE OF THE WAY SOLAR ENERGY IS BEING APPLIED—A PLANT FOR CONCENTRATING SUGAR JUICE IN NEW DELHI, INDIA

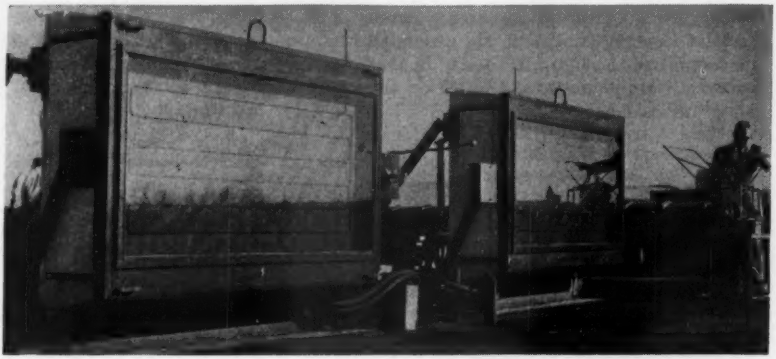
addition, the blending into a structure of the large surface areas required for solar collection, and the adaptation of a suitable solar heat storage system pose architectural challenges.

Once the availability of solar energy has been determined, efficient and economical collection and storage devices have been developed, and integration of the collectors and storage systems into the structures has been accomplished so as to provide an aesthetically pleasing appearance, then the problems of space heating and water heating with solar energy (Sections 5 and 6) are comparatively minor. There are, however, a number of variations in system design which must be considered for most effective operation. Whether solar energy is to be collected and utilized directly; whether it is preferable to reduce the size of collectors and to utilize intermediate energy lifts such as through heat pump systems; whether the systems should be designed with additional auxiliary heat; and whether or not the solar collection should be with air, water, or some other medium as a transport agent are but some of these problems. In addition, domestic hot water heating may be combined with space heating or even cooling. An extension of the use of the solar system to the heating of swimming pools, perhaps both to extend the use of the

pool beyond the normal seasons and also for adaptation of the swimming pool as a solar storage system, indicates additional possibilities.

Comparatively little effort to date has been placed upon the adaptation of solar energy to comfort cooling or to the low-temperature preservation of foods (Section 7—Cooling and Refrigeration). Inherently, such considerations have real appeal and practicality since those areas of the world which have the highest solar availability also have the greater needs for cooling and refrigeration.

Concomitant with the rapid increases in world population and the increasing industrialization of the majority of the world's countries are also the burgeoning demands for pure water (Section 8—Distillation). There are few areas of the world which have sufficient pure water to supply the long-range needs when full



SOLAR COLLECTOR IN USE IN UNIVERSITY OF MINNESOTA'S SOLAR PROGRAM

consideration is given to the requirements of human consumption, agriculture, and industry. Again, the greatest potential needs appear to be in those sections of the world where the availability of solar energy is highest and the availability of potable water is lowest. The extension of agriculture and industry to arid or semi-arid areas depends greatly upon the water supply. Considerable progress has been made in the design of solar stills which utilize solar energy for the evaporation of brackish water and the subsequent condensation of pure water. Where such devices are practical the availability of solar energy is frequently favorable and with solar stills the problems of storage and architecture are minor. In most respects the problem resolves to the design and construction of an economical solar collection and distillation system. The costs of water obtained by standard means vary widely, and in some areas the use of solar distillation devices is imminently practical. There is no present indication, however, that solar evaporation can provide water from the sea for agricultural use at an economically interesting cost.

The problems of solar utilization for drying and concentrating (Section 9) have been only lightly explored. In the case of agricultural products such as vegetables and fruits, it is now impossible to consume the entire production at the time of the

harvesting, and provisions must be made to avoid deterioration either by placing in cold storage or by drying. In the industrialized countries, facilities for cold storage are available, but in the industrially retarded countries these products are dried by direct exposure to sunshine. Such a process is both time-consuming and unhygienic. In places where fuel is scarce or in short supply and plenty of sunshine is available, solar energy could profitably be utilized and the drying time considerably reduced through the application of solar collection and heat transfer devices. The seasoning of wood, the drying of vegetables and fruits, and the preparation of common salt from sea water suggest other applications. Similarly, in such countries as India, cottage and small scale industries can concentrate palm and sugar cane juices directly into jaggery or thick syrup by the use of concentrators now under development.

It is hoped that these few comments together with the accompanying outline will be of aid in orienting the thinking of solar energy utilization problems and in the suggesting of fruitful areas for research. The Technical Advisory Committee on Solar Energy Utilization will attempt to answer specific inquiries directed to the committee through the Society in those cases where individuals or groups of individuals are planning to embark on an organized program of research in solar energy utilization.

ACKNOWLEDGMENT

Acknowledgment is made to those members of the Committee who have arranged for making available the photographs included here, *viz.*: R. C. Jordan, Minneapolis, Minn.; M. L. Khanna, New Delhi, India; J. L. Threlkeld, Minneapolis, Minn.; G. T. Ward, Singapore, Malaya; and L. F. Yissar, Israel.

OUTLINE OF TECHNICAL INFORMATION NEEDED FOR SOLAR ENERGY UTILIZATION

Section 1—Availability of Solar Radiation

1.1 ***Availability during clear days***

- **1.11** Normal incidence
  - **1.111** At various times of year
  - **1.112** At various locations on earth's surface
- **1.12** Incidence of direct solar radiation upon variously oriented surfaces
  - **1.121** Outside the atmosphere
  - **1.122** At various locations on earth's surface
  - **1.123** Hourly rates of incidence at various times of year
- **1.13** Diffuse radiation from sky
  - **1.131** Incidence upon variously oriented surfaces
  - **1.132** Variation of incidence with time of year
- **1.14** Diffuse radiation reflected from ground and other surfaces
  - **1.141** Determination of ground cover reflectances
  - **1.142** Effect of receiving surface orientation
- **1.15** Total radiation
  - **1.151** At various locations on earth's surface
  - **1.152** Hourly rates of incidence at various times of year
  - **1.153** Effect of receiving surface orientation
  - **1.154** Determination of design values of instantaneous radiation rates for various times of year and for variously orientated surfaces

1.2 *Availability during all days*

1.21 Total solar radiation received

1.211 Average hourly rates for days with various percentages of possible sunshine

1.212 Average hourly, daily, weekly, and seasonal totals of direct and total radiation for various locations

1.213 Statistical distribution of clear, cloudy, and partly cloudy days for each month for various locations

1.214 Sequences of clear, partly cloudy, and cloudy days for various locations in combination with outdoor temperature

1.215 Statistical distribution of hours with various levels of total solar energy intensity, for various locations

Section 2—Solar Collectors

2.1 *Energy source*

2.11 Direct solar

2.12 Diffuse and direct solar

2.2 *Geometry of collector design*

2.21 Orientation relative to solar source

2.211 Fixed

2.212 Movable

2.2121 Rotation in one plane

2.2122 Rotation in two planes

2.2123 Seasonal adjustments

2.22 Solar concentration by lens focusing

2.23 Solar concentration by reflection

2.231 Reflection in two dimensions

2.2311 Parabolic trough (line focus), circular trough

2.2312 Truncated triangular trough (plane focus)

2.2313 Hybrid (earth, plaster, multiple mirrors, etc.)

2.232 Reflection in three dimensions

2.2321 Paraboloid (point focus), hemisphere

2.2322 Cone (line focus)

2.2323 Hybrid (earth, plaster, multiple mirrors, etc.)

2.233 Albedo

2.24 Plane collectors

2.241 Box

2.242 Flat plate

2.3 *Temperature level of collection*

2.31 High temperature (furnace)

2.32 Intermediate temperature (heat collection for processing and cooking, refrigeration, space heating and cooling, power)

2.33 Low temperature (space heating, water heating, heat pump-heat source)

2.4 *Transluscent solar trapping materials*

2.41 Material

2.411 Glass

2.412 Plastics

2.413 Other, *i.e.*, oil-film, chemical coatings

2.42 Radiation properties (reflectivity, absorptivity, transmissivity) for both solar and low temperature sources

2.43 Physical properties, *i.e.*, durability

2.44 Number of independent layers or traps (single, multiple)

2.5 *Absorbing surfaces*

2.51 Radiation properties

2.511 Emissivity (solar and low temperature)

2.512 Discrimination

2.5121 Wave length

2.5122 Wave front

2.52 Geometry

2.521 Relationship between transport medium and absorbing surface

2.5211 Medium contained in absorbing surface (double sheet, sheet and tube, etc.)

2.5212 Medium external to absorbing surfaces (gauze, parallel glass plates, etc.)

2.5213 Configuration of tubing or passages

2.6 *Reflecting surfaces*

2.61 Materials for specular layer

2.62 Materials for specular layer support

2.63 Materials for specular layer protection

2.64 Assembly methods

2.7 *Transport media*

2.71 Water

2.72 Air

2.73 Other

2.74 Multiple media

Section 3—Solar Heat storage

3.1 *Means of Storage*

3.11 Specific heat (rock, earth, water, etc.)

3.12 Solid-liquid phase change

3.13 Liquid-gas phase change

3.14 Reversible chemical reaction storage

3.2 *Container*

3.21 Type (tank, bin, earth, part of structure, etc.)

3.22 Location (above or below ground, in or out of structure, etc.)

3.23 Insulation

3.3 *Heat transfer*

3.31 Temperature level

3.32 Rate of heat accumulation and dissipation

3.33 Transfer areas, configuration

Section 4—Architectural Considerations

4.1 *Local conditions*

4.11 Obstructing effect of
4.111 Mountains, hills
4.112 Woods, trees, shrubs
4.113 Surrounding buildings
4.114 Neighborhood taste

4.12 Micrometeorological variations and cloudiness caused by
4.121 Mountains, hills and valleys
4.122 Nearby lakes
4.123 Obstacles surrounding the structure (trees, walls, etc.)

4.13 The ground
4.131 Level ground
4.132 Ground sloping in different directions
4.133 Nature of soil

4.2 ***Types of structures (homes, schools, stores, etc.) adaptable to solar heating and/or cooling, and their characteristics***

4.3 *Windows*

4.31 Daily and yearly heat gains
4.311 Direct-solar with different orientation and inclination (single and multiple pane)
4.312 Reflection or albedo effects
4.313 Heat interchange due to temperature difference with different orientation and inclination (single and multiple pane)

4.32 Protection against direct-solar heat gain
4.321 External protection (overhangs, brise-soleils, awnings, shutters, trees, etc.)
4.322 Internal protection (venetian blinds, drapes, shades, etc.)
4.323 Protection between the panes (venetian blinds, shades, etc.)

4.33 Protection against heat gain or loss due to temperature difference
4.331 External protection (shutters, etc.)
4.332 Internal protection (shutters, drapes, shades, venetian blinds, etc.)
4.333 Protection between the panes (venetian blinds, shades, etc.)

4.34 Thermal properties of glass (transmissivity, reflectivity, absorptivity)

4.4 ***Reception by walls and floors of solar heat through windows***

4.41 Effect of surface material

4.42 Effect of heat conduction by floor and wall material

4.43 Effect of heat capacity of floor and wall

4.5 ***Effect of roof and wall orientation on heat gain or loss***

4.51 Solar heat gain
4.511 Surface material
4.512 Surface color
4.513 Insulation properties
4.514 Ventilated surface construction (heat collector)

4.52 Heat gain and heat loss due to temperature difference
4.521 Insulation properties
4.522 Surface material
4.523 Ventilated surface construction (heat collector)

4.6 ***Optimum solar heat utilization with type and arrangement of structure***

4.61 Single family houses
4.62 Terrace houses (orientation of the row)
4.63 Apartment houses (arrangement of apartments, town planning)
4.64 Special houses (laundries, schools, etc.)

4.7 ***Integration of solar collector***

4.71 Location in building surfaces (roofs, walls, windows)
4.72 Separated collectors
4.721 Attached to building
4.722 Located near building

4.8 ***Integration of solar storage system***

4.81 Storage in building elements, (walls, floors, etc.)
4.82 Storage located inside building
4.83 Storage located external to building

Section 5—Space Heating

5.1 ***Architectural considerations (see Section 4)***

5.2 ***Collectors (see Section 2)***

5.3 ***Heat storage (see Section 3)***

5.4 ***Transport***

5.41 Water
5.411 Piping and insulation
5.412 Pump
5.413 Protection against freezing
5.414 Thermal expansion
5.415 Valves
5.42 Air
5.421 Ducts and insulation
5.422 Blower
5.423 Dampers
5.43 Low boiling point liquids
5.431 Pipe size
5.432 Low boiling point liquid flow control
5.433 Oil circulation
5.434 Valves

5.5 ***Energy conversion***

5.51 Collector transport medium same as fluid circulated for heat distribution
5.52 Secondary transport medium with heat exchange
5.53 Temperature lift with heat pump

5.6 ***Heat utilization***

5.61 Direct space heating transmitting radiation through windows, etc.
5.62 Indirect heating with separate collectors

5.7 ***System design***

5.71 Capacity requirements of structure
5.711 Maximum steady-stage heat loss of structure determining peak capacity
5.712 Transient heat loss under diurnal fluctuations
5.713 Sequence of weather conditions controlling insertion of heat into, or withdrawal from, storage and probable duration of supplementary heating if employed
5.72 Collector design (see Section 2)
5.73 Storage system design (see Section 3)
5.74 Heat distribution system design
5.75 Integration with solar cooling and domestic hot water
5.76 Flow chart of system
5.77 Control sequence and components

5.8 ***Solar heat for outdoor living***

5.81 Heating ground in garden place
5.82 Heating water in swimming pool

5.9 ***Economics***

5.91 Effects of different insulations (windows, walls, roof)
5.92 Effects of glass area to wall ratio

5.93 Effects of auxiliary heating

5.94 Solar heat source heat pump systems

Section 6—Domestic Water Heating

6.1 *Requirements*

6.11 Peak load

6.12 Daily load

6.13 Source and discharge temperatures

6.2 *System design*

6.21 Collector

6.22 Storage

6.23 Gravity versus forced circulation

6.24 Direct and indirect solar heating

6.25 Relationship to building design

6.26 Integration with solar space heating and/or cooling system

6.27 Auxiliary heat source

6.28 Controls

6.3 *Collector design (see Section 2)*

6.4 *Storage design (see Section 3)*

6.5 *Economic factors*

Section 7—Cooling and Refrigeration

7.1 *Design data*

7.11 Comfort air cooling

7.111 Minimum cooling requirements as a function of solar loads for various localities and for various types of structures

7.112 Unit loads *vs.* solar intensity for different localities

7.113 Distribution systems best suited to solar air cooling

7.1131 Lowering temperature of entering air

7.1132 Radiant cooling

7.1133 Evaporative coolers in conjunction with solar draft chimneys

7.12 Refrigeration design data

7.2 *Refrigeration methods*

7.21 Mechanical compression

7.211 Solar power for driving conventional compressors

7.212 Solar (thermal) pumps adapted for refrigerant compression

7.22 Absorption and adsorption

7.221 Absorbate-absorbent combinations suitable for solar absorber temperature limits

7.222 Operation

7.2221 Continuous (circulation of boiling solution)

7.2222 Cycling

7.22221 Day-night

7.22222 Manual

7.22223 Governed by completed desorption

7.23 Expanding air

7.231 Solar (thermal) compressor

7.232 Solar hot air engine-turbine

7.24 Direct conversion of solar energy to refrigeration

7.241 Thermocouples

7.242 Semiconductors

7.3 *Heat Sinks*

7.31 Conventional (*i.e.*, air, water, cooling, etc.)

7.32 Nocturnal radiation

7.4 *Energy storage*

7.41 Storage of heat above ambient temperatures

7.42 Storage of refrigeration below ambient temperatures

7.43 Liquid refrigerant

7.5 *Relationship of refrigeration method and box design*

Section 8—Distillation

8.1 *General*

8.11 Potable water requirements at different locations

8.111 United States

8.112 World, land areas

8.113 Ships

8.114 Emergency rafts

8.12 Cost of potable water at different locations

8.121 Urban

8.122 Rural

8.123 Desert regions

8.124 Emergency

8.13 Rain or dew to supplement distilled water

8.131 Seasonal rainfall
8.132 Seasonal dew precipitation
8.14 Geographical areas where solar radiation can be used for distillation
8.141 For total water needs
8.142 With supplemental rainfall and dew collecting devices

8.2 ***Solar distillers with separate heat collectors***

8.21 Characteristics of collectors (see Section 2)
8.22 Evaporators
8.221 Single effect
8.222 Multiple effect
8.223 Flash-still
8.224 Jet-compression still
8.225 Optimum number of stages
8.226 Vapor compression still

8.3 ***Solar stills combining heat collection and evaporation***

8.31 Roof-type stills with tray evaporators
8.311 Effect of roof tilt and shape
8.312 Use of additional reflectors
8.313 Distilled water collecting channels
8.314 Construction of features and materials
8.32 Flat tilted solar stills
8.321 Effect of tilt
8.322 Porous evaporators
8.323 Rigid evaporators
8.324 Construction features and materials
8.33 Multiple effect solar stills
8.331 Effect of number of stages
8.332 Construction features and materials

8.4 ***Cost estimate of solar stills***

8.41 Initial investment costs
8.42 Site preparation and land cost
8.43 Operational costs
8.44 Comparison with other source costs

8.5 ***Special requirements of very small solar stills for remote areas (single families, domesticated animals, etc.)***

Section 9—Drying and Concentrating

9.1 ***Direct heating***

9.11 Direct exposure with natural convection and drying
9.111 Wind velocity, temperature and humidity of different instants and during different intervals
9.112 Properties of drying object (absorptivity, thermal diffusivity, vapor diffusivity)
9.12 Direct exposure with other aids to hasten drying
9.121 Use of glass and plastic films
9.122 Forced convection
9.123 Blackening of container

9.2 ***Indirect heating***

9.21 Medium air
9.211 Blowing air heated directly in a solar absorber
9.212 Blowing air heated by stored solar energy

9.3 ***Concentrating of dilute solutions***

9.31 Direct exposure with natural convection
9.32 Heating containers with energy concentrators (attached or separate)
9.33 Reactivation of sorbents

DISCUSSION

C. P. DAVIS, JR.,† Manhattan, Kansas (WRITTEN): The TAC on Solar Energy Utilization has in broad and nonexclusive terms herein outlined and identified areas of significant technical information necessary to the furtherance of use of solar energy.

To those concerned with explorations in adaptation of this energy source for heating, cooling, and drying in agricultural processing, techniques, and problems, (a small seg-

† Kansas State College.

ment of the larger field) this organization of ideas seems apropos. It no doubt likewise appropriately applies more universally.

The basic studies requisite to the determination of the quantitative availability of the solar source, of the type and magnitude of the storage media, of the collection devices, and the ever present concern with architectural integration, are placed in proper perspective at the forefront of the outline. Understanding of these are fundamental to the development of systems which may exploit the marginal areas of present day feasibility.

This is a noteworthy organization for objective research. Cutting across the scientific and technical disciplines as it does, this outline should assist materially in integrating the contributions each may make.

R. W. SAGE, Linden, N. J.: Can the author give a concept of the amount of solar energy received by the earth in terms of, say, the energy requirements of the U. S? In other words, how much area would be required if solar energy had to be used to supply the energy requirements of the U. S.? This is probably a difficult question because of qualifications such as geographical location and hours of sunlight. I know, though, that some thought has been applied to it, and I am curious to know what the numbers look like.

AUTHORS' CLOSURE (Dr. Jordan): The amount of solar energy received by the earth each day is far in excess of the world's needs if it could be harnassed. Earlier in this decade the United States used approximately 164,000 kilocalories of fuel energy or 186 kilowatt hours of heat per person per day. At the same time the energy striking the land area of the United States was approximately 313,000 kilowatt hours per person per day.

This comparison is misleading, however, since solar energy is very diffusely spread and its collection requires the construction of expensive absorbers covering the areas of collection involved. Thus it is only practical to treat limited areas with such solar collectors. There would also be the real possibility of serious meteorological and climatological disturbances resulting from any material reduction in the total amount of solar energy which reached the earth as heat.

Perhaps a more practical comparison can be made by assuming that the roof of a house with about 1,000 sq ft photovoltaic cells with an efficiency averaging 10 percent over an 8-hr sunshine day. This would provide somewhat better than 2¼ kilowatts of electrical energy averaged over a 24-hr day. Such a system would be capable of supplying the majority, if not all, of the electrical energy needs of the average household and a 10 percent efficiency is already obtainable under good conditions. However, the expense of such a solar conversion system is much too high at the present time for practical consideration.

In Memoriam 1958

Name	Joined
William Allen*, Milwaukee, Wis.	1938
Harold L. Alt, (*Life Member*), West Palm Beach, Fla.	1913
Edward E. Ashley, Jr., (*Life Member*), New York, N. Y.	1912
Raymond P. Babcock, Newton, Conn.	1948
James V. Bamford, Los Angeles, Calif.	1956
Joseph Bastow, Jr., Pittsburgh, Pa.	1958
Reiner P. Bon, Milwaukee, Wis.	1950
Leonard W. Bonkemeyer, Greensboro, N. C.	1957
Arthur F. Bowers, (*Life Member*), Milwaukee, Wis.	1919
Carlos E. Bronson, (*Life Member*), Kewanee, Ill.	1919
Carl E. Brown, Ontario, Calif.	1955
Simon Z. Burk, Norfolk, Va.	1952
Jeremiah E. Carey, Westfield, Mass.	1955
L. O. Ray Clark, Hartford, Conn.	1944
Thomas E. Crone*, (*Life Member*), New York, N. Y.	1920
Henry W. Dirks, Toledo, Ohio	1954
Don M. Ferris, Detroit, Mich.	1944
Frederick S. Frambach, Westfield, N. J.	1946
Joshua L. Garvin, Charleston Heights, S. C.	1945
James Gayner*, San Francisco, Calif.	1937
Leslie S. Gilbert, Dallas, Tex.	1937
Robert H. Grossman, Denver, Colo.	1950
Fred E. Grosvold, Eau Claire, Wis.	1928
Earl L. Hilburg, Los Angeles, Calif.	1955
Edwin C. Hodge*, Fort Worth, Tex.	1957
Edward L. Hogan, (*Life Member*), Clearwater, Fla.	1911
Glenn Holford*, Decatur, Ga.	1949
Carl W. Johnson, (*Life Member*), Chicago, Ill.	1912
Philip B. Johnson†, Columbus, Ohio	1945
James J. Joyce, Atlanta, Ga.	1941
Peter J. Keegan, Flushing, N. Y.	1949
Dwight D. Kimball, (*Presidential and Life Member*), New York, N. Y.	1908
Andrew W. Knecht, III, New York, N. Y.	1957
Charles H. Koper, Cincinnati, Ohio	1951
Daniel H. Lewis, Detroit, Mich.	1944
Gustav Frederick Linck, Baltimore, Md.	1949
Rowland H. Lock, Toronto, Ont., Canada	1939
Samuel E. Lyman, (*Life Member*), New York, N. Y.	1924
Bernard Lyons*, Springfield, N. J.	1954
Albert B. Martin, (*Life Member*), Iowa City, Iowa	1916

* Died in 1957, notification received 1958.
† Died prior to this year, but notification received in 1958.

In Memoriam 1958 *(continued)*

Julien W. Mathis, (*Life Member*), Chicago, Ill.	1921
Carl R. Matthews, Westlake, Ohio	1944
Howard F. McCandless, Fresno, Calif.	1946
Laban Jack, McCune, Tulsa, Okla.	1942
Alfred E. McGruer, Montreal, Que., Canada	1945
Charles P. Merryman*, Union City, Tenn.	1945
H. Preston Morehouse, Short Hills, N. J.	1933
Frank X. Morgan, Philadelphia, Pa.	1947
Laurence K. Nelson*, New Orleans, La.	1940
Robert W. Nelson, Louisville, Ky.	1950
Victor E. Nelson, Camden, N. J.	1950
James C. Niemeyer, St. Paul, Minn.	1949
Ernest H. Norton, Seattle, Wash.	1946
John A. Norton, Leaside, Ont., Canada	1940
M. Earl Ott, Los Angeles, Calif.	1942
Jeff Davis Owen, Downey, Calif.	1937
Charles J. Peters, Baltimore, Md.	1946
Charles M. Pierson, Bloomfield, Ill.	1954
Alexander Pirrie, Toronto, Ont., Canada	1955
C. Wesley Potter, Lead, S. D.	1949
Robert W. Powers, Bennettsville, S. C.	1941
Van A. Reed, Jr., (*Life Member*), Elizabeth, Pa.	1930
Charles A. Reif*, Buffalo, New York	1937
Arthur Ritter, (*Life Member*), Scarsdale, N. Y.	1911
Louis Schwartz*, St. Louis, Mo.	1953
Cyril D. Seymour, Willowdale, Ont., Canada	1945
Irwin D. Shea, Canoga Park, Calif.	1950
Edwin B. Sheffield, Toronto, Ont., Canada	1921
Robert E. Sinclair, Portland, Ore.	1952
J. Walter Singmaster*, Allentown, Pa.	1943
Lawrence C. Soule*, (*Life Member*), Montclair, N. J.	1908
Charles H. Speckman, (*Life Member*), Philadelphia, Pa.	1918
Ray L. Spitzley, (*Life Member*), Detroit, Mich.	1920
Edward L. Stammer, (*Life Member*), St. Louis, Mo.	1919
Charles Stevens, Los Angeles, Calif.	1951
Harold J. Taylor, Detroit, Mich.	1936
Harold A. Thornburg, New York, N. Y.	1929
Robert K. Thulman, Washington, D. C.	1938
Mervyn G. Walters, San Fernando, Calif.	1950
Dick W. Whittington, Indianapolis, Ind.	1949
Harvey O. Williams, Houston, Tex.	1945
Ray A. Wise*, Cleveland, Ohio	1944

* Died in 1957, notification received 1958.

Dwight D. Kimball

1874-1958

Dwight D. Kimball, retired consulting engineer who served as twenty-second president of the Society in 1915, died in his New York apartment on January 11th, 1958. Mr. Kimball, who was 83 years old, had been failing in health for some time.

When he retired a few years ago, Mr. Kimball had an outstanding record of engineering accomplishments. During his active years he planned the heating, ventilating and air conditioning for many of the outstanding buildings in the country, including his work on the heating and ventilating system for Madison Square Garden in New York when it was built in 1925. Others of his projects embraced the state capitols at Jefferson City, Mo., and Olympia, Wash., and buildings on the Cornell, Princeton and Yale University campuses. His engineering work extended to South America and Europe.

Mr. Kimball, who was born in Dover, N. H., was graduated from Dartmouth College and started his lengthy career with the R. D. Kimball Co., Boston, Mass., in 1895. Three decades later he opened his own office as a consulting engineer in New York, N. Y. He entered into partnerships with Messrs. Cucci and Henszey at different intervals in the ensuing years. Besides his business activities, he was a lecturer at Columbia University.

In his fifty years of Society membership, Mr. Kimball generously contributed his time and effort. In 1912–13, just four years after becoming a member, he served on the Board of Governors of the Society. He was elected 2nd vice president for 1914 and president for the following year. In 1914 and 1916 he was a member of the Council, presiding as chairman in 1915. Throughout the years, he served on the Advisory Board.

Mr. Kimball was chairman of the New York Chapter, 1911–12, and later was a member of the Entertainment Committee, New York Chapter, which then acted as host for the Society. His wide range of committee activities included membership on the Schoolroom Ventilation Committee, 1913; Finance Committee (of Council), 1916; Committee on Revision of Constitution, 1922–25, and Committee to Confer with *American Institute of Architects*, 1923–25. In 1923 he was active also on

the Committee to Consider the Report of the New York State Commission on Ventilation, and was chairman of the subcommittee on Direct Steam or Hot Water Radiation. His interest in ventilation was exhibited by membership on the Committee to Develop a Ventilation Safety Code under the Procedure of the American Engineering Standards Committee, 1925, and the Committee on Ventilation Standards, 1931–34.

As an author, Mr. Kimball contributed technical papers which appeared in the TRANSACTIONS of the Society. He wrote on Ventilation Problems and was joint author of presentations on the Experimental Laboratory of the New York State Commission on Ventilation and a Description of the First Year's Work, and on The Testing of Atmospheric Conditions and Heating and Ventilating Equipment.

Surviving are his wife, Mrs. Blanche Kimball, and a brother, John V. Kimball.

INDEX

A

B

C

D

E

F

G

H

I

J

K

L

M

N

O

P

Q

R

S

T

U

V

W

Z